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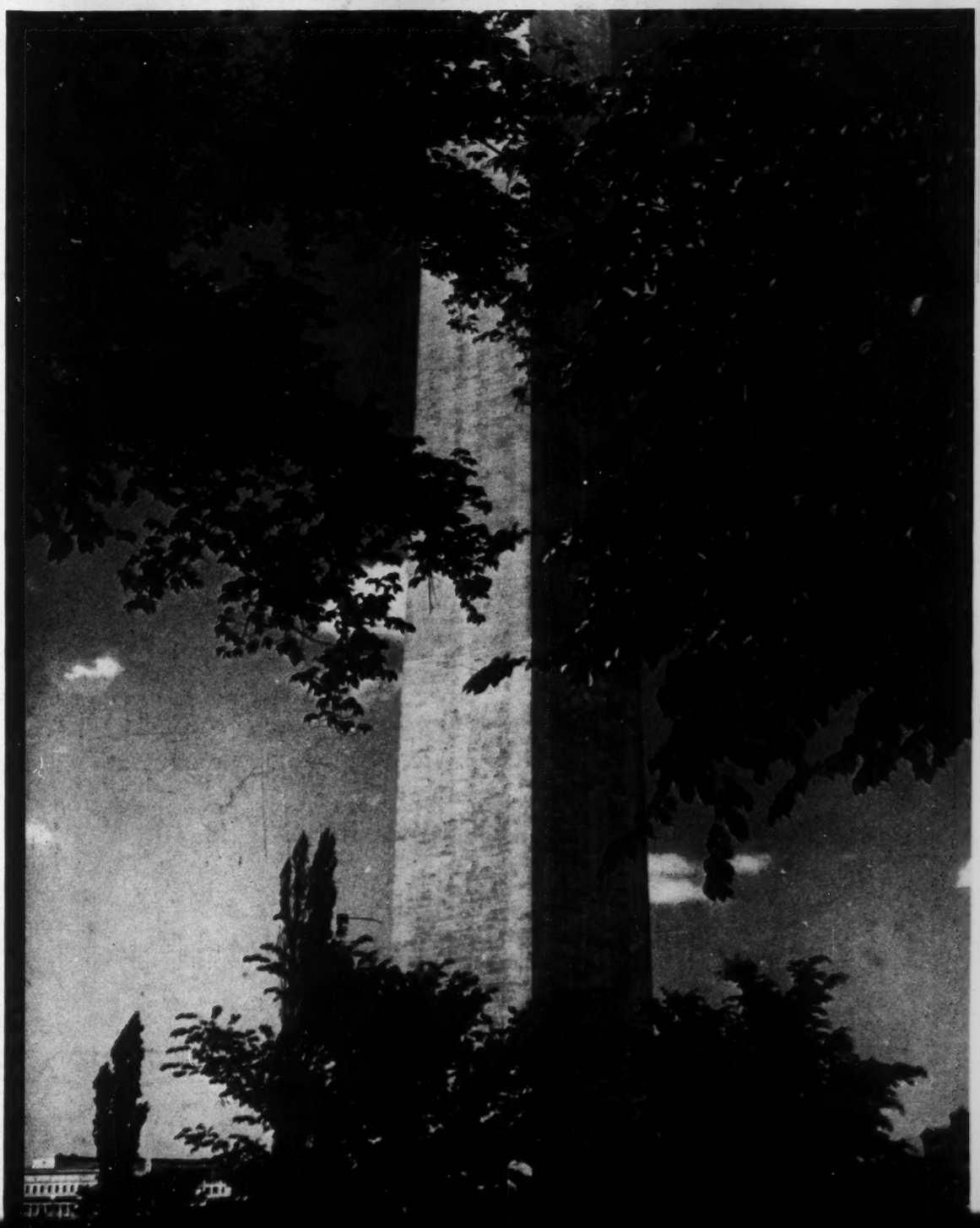

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Page 371
Journal Section
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Texrope Helps Bring Comfort to Movieland Spa!

SHORT-CENTERED, COMPACT VARI-pitch drives like those at Arrowhead are simply and quickly adjusted to control fan operation . . . their stationary control features being adapted to infrequent changes. Heating, ventilating and air-conditioning contractor for Arrowhead was W. S. Kilpatrick & Co.; general contractor, Wm. Simpson Construction Co.; consulting engineers, Holmes & Narver; architects, Gordon Kaufmann and Paul R. Williams.

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The Editor's Pages

OPM CONSULTANT LOOKS TO FUTURE

Nondefense industries must plan for "a period not of months but of years" to get along with much less than their requirements in a long list of strategic materials, Philip D. Reed, chairman of the board of the General Electric Co., said at the annual meeting of the National Industrial Conference Board last month.

Mr. Reed, who for three months has been senior consultant to the director of priorities of the Office of Production Management in Washington, said he was expressing "a few personal observations on several points that must be considered in charting the course of any business."

Prime emphasis will be placed on military products for five or more years, he said, and added that "the sooner we accept the fact, for purposes of planning, that we face a long period of enormous production for defense, with a consequent shortage, rationing and allocation of strategic materials, the more quickly will nondefense industries adjust themselves to the new conditions and undertake the great task of maintaining maximum production for civilian needs without interference with defense output." He suggested "the redesigning of peacetime products wherever possible, and where impossible the preparation for defense manufacture" to prevent shutdowns and unemployment.

Mr. Reed predicted that engineers, chemists and designers will develop new products to help absorb the increased purchasing power, allay inflationary pressures, support our standard of living, and keep the country at work. One of the benefits of a larger national income, he said, would be an increased ability to pay for a large part of the defense program as we go.

Under present conditions, Mr. Reed said, "unnecessary accumulation of scarce materials, in raw, fabricated or scrap form, either by defense or nondefense manufacturers or dealers, is seriously hurtful to the national program and must be recognized as undesirable, unpatriotic and ultimately profitless. I am confident that no more is required than to call this matter to the attention of industry and business, as we are now doing."

The post war decade will resemble neither the 1920's nor the 1930's, said Mr. Reed. He summed up the following points to be expected in the future:

A mature, consolidated, national labor organization, led by able and understanding men, who will negotiate all questions pertaining to wages, hours, and conditions of work, with business leaders acting in concert through associations of employers on an industry basis.

A continuation of the trend toward centralized control of money and credit.

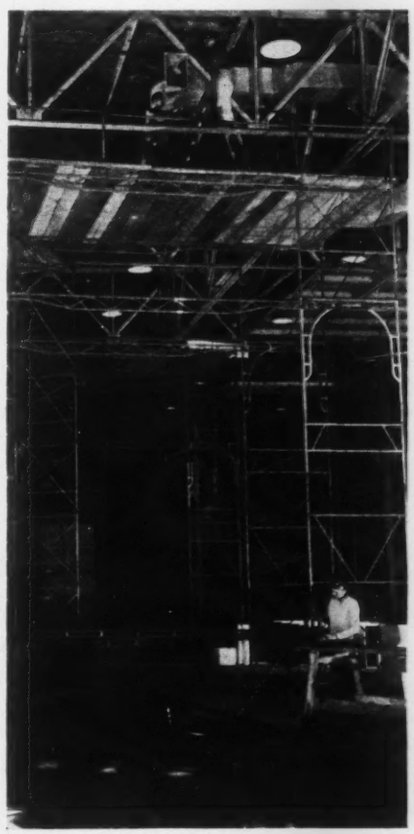
Large and gradually declining government spending for previously planned and generally desirable public projects, for several years after the war, in order to cushion the devastating effect of an almost vertically declining demand for armaments.

Great emphasis placed on production and a deep determination to take whatever steps are necessary, however drastic, to avoid a recurrence of mass unemployment. To this end we may expect to see the government gradually relinquishing its ownership of defense plant facilities on terms and conditions that will provide the purchaser with a strong incentive to maintain production and employment.

Subject to these changes, a sincere attempt to preserve the elements and incentives of the enterprise system.

Whether under these conditions, and with profit margins generally lower than in the past, there will be enough confidence to create a broad flow of private savings into new and existing enterprises will be the factor that determines whether more drastic experiments in government borrowing and spending will be resorted to.

Night crews installing air conditioning ducts in one of the buildings of the Douglas Aircraft Co.'s huge new plant at Long Beach, Calif. A 4000 ton decentralized air conditioning system serves this windowless black-out plant, as described on p. 307 of the May HPAC



NEW CHAIRMAN FOR ARMY CONSTRUCTION GROUP

Major-general William D. Connor, USA retired, has been called to active duty, and assigned as chairman of the construction advisory committee of the office of the quartermaster general, the war department has announced. The department also announced the appointment of Dean James H. Graham, of the engineering school, University of Kentucky, and formerly a colonel in the army, as technical advisor to the chief of the construction division, office of the quartermaster gen-

The Priorities Critical List—A Classified Summary*

(This is not a complete enumeration. The number of items classified under each heading is shown in total, but only those items of broad importance or special interest are listed)

¶Raw and Semi-Manufactured Materials also Having Peace-time Uses (20 items)
Aluminum
Ammonia, anhydrous
Brass
Bronze
Chlorine
Chrome Steel
Cordage
Lenses (special)
Magnesium
Monel Metal
Nickel
Oil: castor and special diesel fuel
Potassium perchlorate
Rubber, synthetic
Steel: various types of special alloys (but not all kinds of steel)
Tin
Tungsten
Vanadium
Zinc

¶Machinery and Manufactured Goods also Having Civilian and Industrial Uses (58 items)
Batteries: radio, ship, and fire control
Bearings: roller and ball
Binoculars (and related optical devices)
Cable, electric (various special types only)
Cameras: aeronautical, gun, triangulation, motion picture, and still
Cranes
Electric generators, motors, and motor generators (all except NEMA standard types)
Locomotives: diesel, gasoline, and electric
Machine and metal-working tools
Pumping sets, all types
Radio apparatus, all types
Searchlights and control instruments
Tools, precision: hand
Telephone: special types
Watches: special types
Wires: special types

¶Aircraft and Equipment (13 items)
Aircraft: all types complete
Balloons: observation or barrage
Catapults
Flying Equipment
Fuel
Parachutes
Training Devices

¶Ordnance (50 items)
Chemical warfare material
Explosives
Fire control instruments and equipment
Gas: decontamination apparatus and masks
Guns: all types and calibers (including carriages and mounts)
Ammunition: small and large all types
Armor plate
Mines: land and sea types
Nets: anti-submarine and camouflage
Rifles: automatic and semi-automatic
Small Arms: all types and calibers
Torpedoes

¶Ships (13 items)
Barges
Boats (numerous types listed)
Ship plates
Ships: all types complete

¶Motor Vehicles (7 items)
Ambulances
Combat Vehicles
Motorcycles
Tanks
Tractors
Trailers
Trucks

¶Construction Other than Ships (6 items)
Boilers: power plant and heating
Bridges: portable, steel, and pontoons
Drydocks: floating

¶Personal Equipment Requirements of Armed Forces (9 items)
Caps: field and winter
Clothing: flying and chemical protective
Cotton: canvas, duck, and webbing
Identification tags

*Abstract from the list released by the Priorities Division, Office of Production Management, March 15, 1941, and published in booklet The War of Metals issued by Dun & Bradstreet, Inc.

eral. Both General Connor and Dean Graham already have assumed their new duties.

Appointment of General Connor brings the membership of the construction advisory committee to six.

ACRMA ADOPTS STANDARDS AND ELECTS OFFICERS

The Air Conditioning and Refrigerating Machinery Association, Inc., Southern Bldg., Washington, D. C., in its meetings at Hot Springs, Va., last month, approved and adopted six important equipment standards. These cover self-contained cooling air conditioning units; self-contained room coolers and self-contained room air conditioners; "Freon-12" evaporative condenser units; horizontal closed shell and tube "Freon-12" condensers; "Freon-12" liquid receivers; and location and inspection of data plate on insulated refrigerant containing vessels. This makes a total of 25 equipment standards which ACRMA and its predecessor associations have formulated and adopted in the past three years.

E. T. Murphy of Carrier Corp. was elected ACRMA president for the coming year, to be assisted by C. E. Wilson of Worthington Pump and Machinery Corp. as first vice-president, J. P. Rainbault of General Electric Co. as second vice-president; and P. A. McKittrick of Parks-Cramer Co. as treasurer.

W. H. Aubrey of Frick Co. was elected to the chairmanship of the ACRMA board of directors. In addition to the officers, the members of the board are: P. Y. Danley, Westinghouse Electric & Mfg. Co.; J. M. Fernald, Baker Ice Machine Co.; F. T. Goes, Vilter Mfg. Co.; G. A. Heuser, Henry Vogt Machine Co.; S. E. Lauer, York Ice Machinery Corp; D. W. Russell, Airtemp Div., Chrysler Corp.; H. R. Sewell, B. F. Sturtevant Co.; and G. E. Wallis, Creamery Package Mfg. Co. William B. Henderson continues as executive vice-president.

CORE MAKER'S HELPER TO PRESIDENT

The board of directors of Crane Co. elected John H. Collier as president last month to fill the vacancy created by the sudden death of Charles B. Nolte on April 29. Mr. Collier was born in Chicago in 1884, was educated at Purdue University, and joined Crane Co. on leaving school in 1903. From his original position as core maker's helper 38 years ago he has progressed through a variety of positions, giving him an intimate and thorough grasp of the company's activities.

Silver Anniversary—Bell & Gossett Co., Chicago, celebrated its 25th anniversary with a sales meeting May 7, 8 and 9, attended by approximately 100 of its representatives from all sections of the country. High spot of the meeting was a banquet the evening of May 8 at which company officials reported breaking ground for a new factory, greatly increased sales figures.

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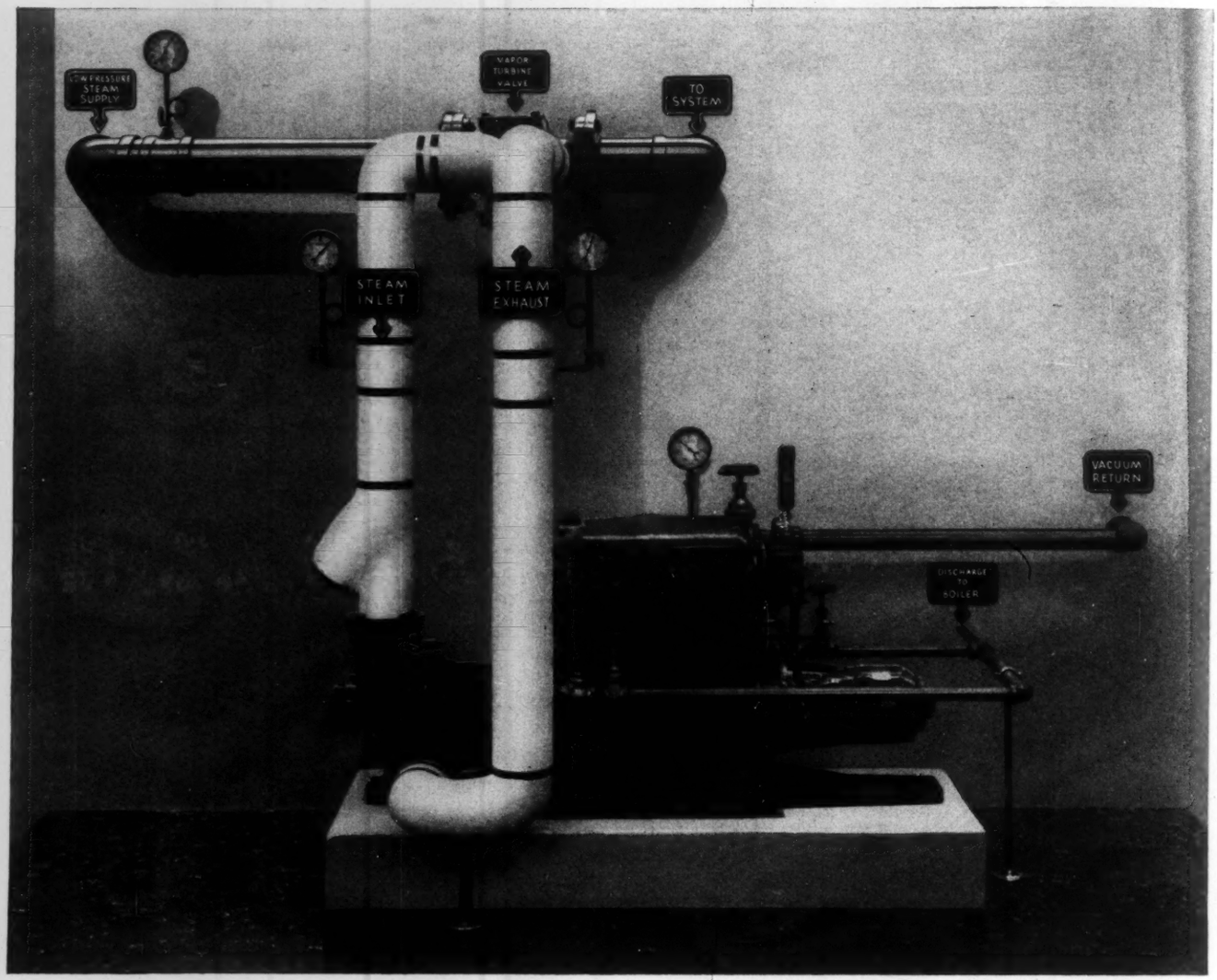
Where Heat Must Not Fail— Install this SAFE Heating Pump



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump. Bulletin on request.

THE NASH ENGINEERING COMPANY

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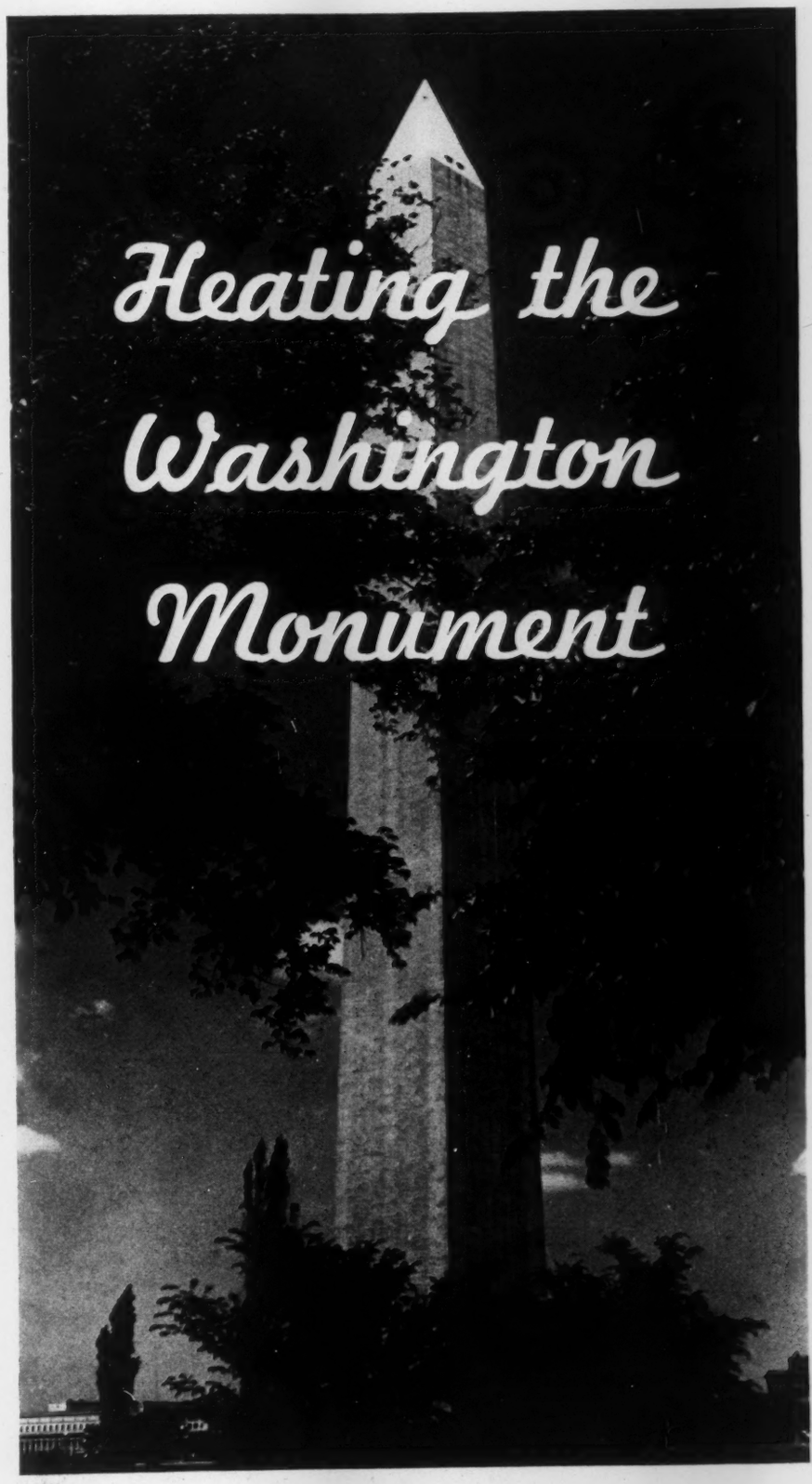
Heating the Washington Monument



CHARLES S. STOCK TELLS HOW SHAFT IS HEATED TO PROTECT STEEL WORK FROM DAMAGE BY CONDENSATION . . . FIGURING REQUIREMENTS A TRICKY JOB

MONUMENT—Ever since a contract was awarded last year for a heating installation in the Washington national monument there's been speculation and conversation about it; some have wondered why—after all these years—a heating system was deemed necessary. In the first description of this unusual project, Charles S. Stock, district representative of the Herman Nelson Corp., tells why heating is necessary to protect the steel work from damage by condensation—government engineers have realized for some time that steps would have to be taken to halt deterioration of the structure. In addition, heating will provide more comfortable conditions for the thousands who visit the monument every year and will make for safer conditions by eliminating slippery stairways. . . . . How to figure the required heating capacity of the units was a tricky question, and Mr. Stock explains the reasoning behind and the basis of the calculations. It was decided to handle the job by installing unit heaters distributed throughout the lower 100 ft of the 555 ft shaft

FOR over a decade there has been knowledge on the part of government engineers that steps would have to be taken to preserve the steel work in the interior of the 555 ft Washington national monument. Last year a contract for a heating system was awarded to S. E. Dockstader. The primary purpose of the system is to eliminate excessive and damaging condensation, which was causing the steel work within the monument to disintegrate. The condensation occurs mostly in summer due to warm outside air entering the monument and being chilled by the cold mass of the structure.

This steel work supports the stairs and elevator shaft and its preservation is a matter of economy, for the difficulty and cost of renewing it would be far greater than the cost of a heating system designed to preserve it. That this steel work is put to considerable use by visitors to this national shrine is undeniable. As many as 10,250 persons a day have ascended the stairs and elevators, and as many as 4800 have ascended the stairs alone. In addition to damaging the steel work, the condensation caused the stairs to become dangerously slippery. Thus, the heating of the monument was a safety precaution as well as a matter of economy.

The monument has been disagreeably cold and damp and the

heating system will also be a worthwhile addition for the improvement of comfort.

Unusual Problem of Design

From the technical standpoint the design of the system presented a problem subject to many limitations and one that was without precedent or data upon which to proceed. Two methods were available for eliminating the trouble caused by condensation. One was to lower the dew point temperature of the air within the monument below the temperature of the structure and the other to raise the temperature of the structure above the dew point temperature of the air within it.

It would not be possible to calculate accurately the quantity of heat necessary to compensate for the heat loss of the monument. If direct radiation and accepted methods of calculation were used, it would theoretically be necessary to maintain a temperature of nearly 450 deg at the top of the shaft in order to maintain a temperature of 40 deg at the base. In addition, the updraft due to stack effect, when established by the heating system, would cause a quantity of unheated air to enter the shaft at the base, which quantity could not be accurately determined. During the summer, the air that could infiltrate into the shaft would be limited to the quantity that could escape through the open windows at the top of the shaft. If it were assumed that the air passed through the windows at a high velocity, say 800 to 1000 fpm, the quantity of escaping air would be equal to about 4½ to 5½ changes per hour. Such a quantity, however, could not escape from the top unless it could enter at the bottom. Because a considerably smaller quantity would enter at the bottom, being limited to that which could pass through the cracks around the elevator shaft door and through the intermittently opened door to the stairway, and because the heating load in the summer would be light and ultimately probably *nil*, it is logical to assume a considerably smaller quantity of infiltration. On the other hand, if the windows were closed, as would be the case during the winter, the quantity of air that could escape through the cracks around the sash would be equal to about ¼ of an air change per hour, assuming an amply high velocity of 1000 fpm. In view of the above noted extremes, about 1½ changes would be a reasonable assumption.

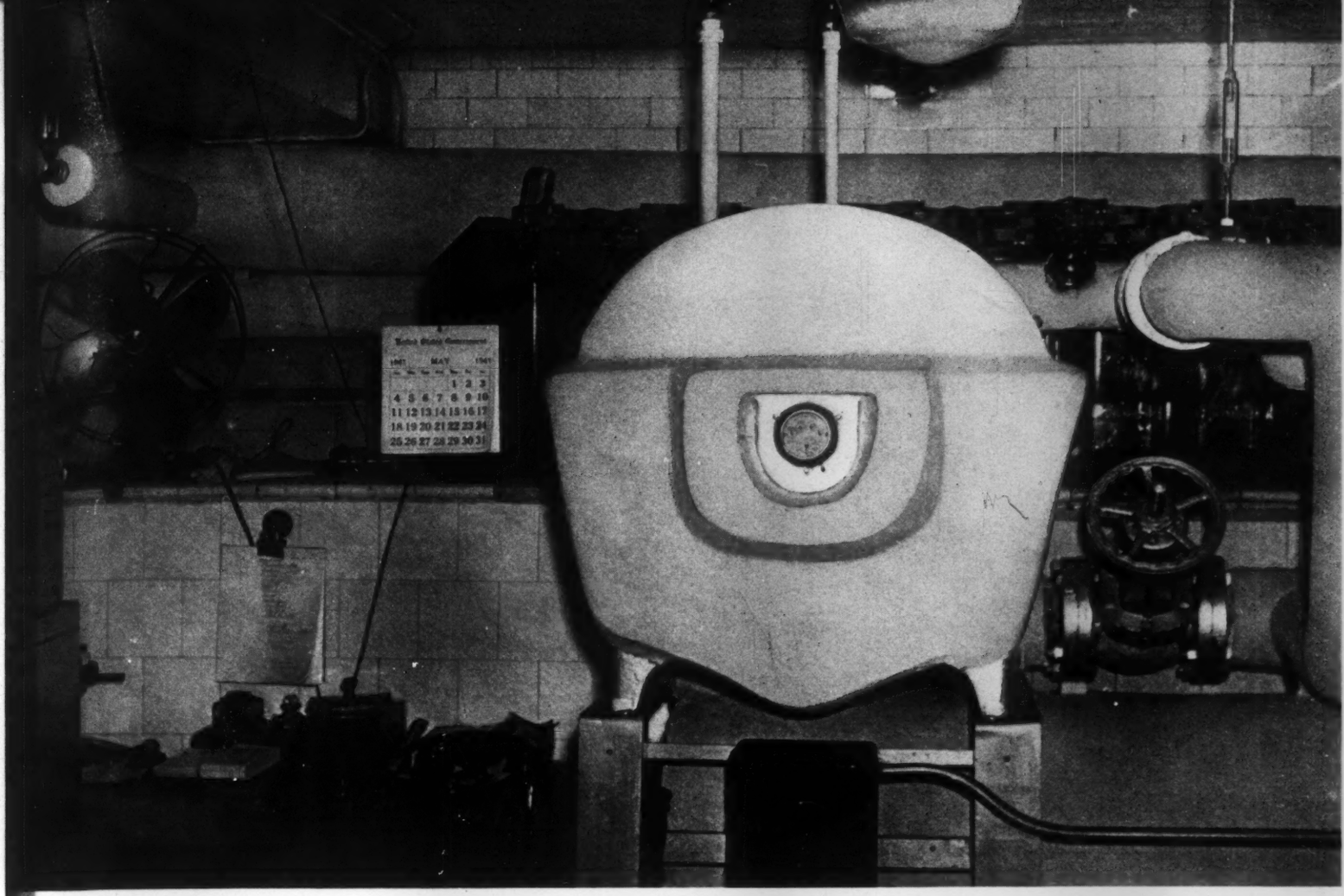
Figuring Heat Loss

In order to secure a tentative figure, the heat loss of the monument was first calculated on the assumption that it was a normal building, neglecting the temperature increasing effect of its excessively high ceiling, and allowing 1½ air changes for infiltration. On that basis, the quantity of heat necessary to maintain an air temperature of 40 deg within the monument was computed to be 1,650,000 Btu per hr.

In order to approximate the degree of inadequacy of this quantity of heat, an additional computation was made to determine the quantity of heat that would be required to raise the temperature of the structure itself. The temperature range through which the structure should be heated in order to raise its temperature above the dew point of the air was somewhat problematical because of the absence of an all-year temperature record of the stone work. It is known, however, that the stone work remains at a low temperature during the summer, because of the fact that heavy condensation within the monument does not cease until near the end of the summer. A range of 30 deg was assumed, being a value that would be ample for conditions at any and all seasons. Proceeding on that basis, it was calculated that approximately 61¾ *billion Btu* would be required to raise the temperature of the stone work—which quantity does not include the heat that would be required to raise the temperature of the steel work, nor that required to maintain the stone work at the increased temperature.

If a quantity of heat equal to the tentatively computed heat loss value were introduced into the monument, computations indicate that it would

The top of the steam main at the 100 ft level in the monument, showing one type of unit heater used. It has vertical discharge louvers so arranged as to deflect the heated air toward the masonry walls



A view of the guard room (which once housed the steam engine which operated the elevator) showing the steam mains entering the monument through the old steam elevator drive shaft opening. The condensate meter is seen in the foreground, and in the background is the emergency storage battery installation

require more than 4½ yr, 24 hr per day, to raise the temperature of the stone work (neglecting the maintaining of the increased stone work temperature and neglecting the increasing of the steel work temperature).

The Design Basis Used

In view of the foregoing, it was obviously logical to proceed on the basis of supplying as much heat to the monument as was practicable. In order to conform to the main limitations, and with the funds available, the maximum practicable quantity of heat to be introduced into the shaft at desirable conditions was established as 2,600,000 Btu per hr, plus the additional heat required for the adjacent guard room. The system was designed on this basis.

Even with that quantity of heat, the computations indicated that some 2¾ yr would be required to raise the temperature of the stone work alone. Therefore, the system was designed, except for certain features, so that additional heaters can if necessary be provided at a future time and so that a maximum possible heat quantity of about 4,400,000 Btu per hr can be introduced into the shaft.

Unit Heaters Chosen

Although direct radiation would possibly have been the better choice for heating the stone work effectively, it was decided to use propeller fan unit heaters for various structural, architectural, physical and aesthetic reasons. The use of unit heaters was considered to have several advantages, such as a considerable reduction in the number of units required, the practicability of a greater flexibility of control and a tendency to resist an updraft due to stack effect by virtue of the fact that each of the 16 unit heaters recirculates several thousand cubic feet of air per minute from and at its own level.

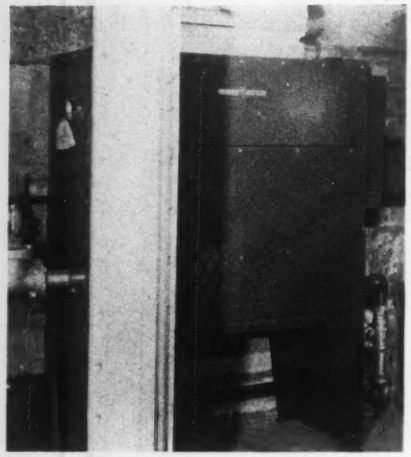
The design provides for the unit heaters to be distributed throughout the lower 100 ft of the shaft; physical and mechanical considerations would preclude the installation of units throughout the entire height of the shaft. Because of the inadvisability of unnecessarily adding considerable weight to the existing steel work, the steam and return risers are so installed that their weight is carried by the foundation of the monument. The temperature to which the air within the monument is to be heated was confined to a minimum value consistent with available commercial equipment, in order that possible strains and stresses would not be set up in the structure. The steam supply riser was insulated for several reasons—principally to prevent the possibility of one column of the elevator shaft from being subjected to a temperature a great deal above that to which the other columns will be subjected.

Control

As will be apparent from the foregoing discussion it is not pos-

sible to anticipate accurately the conditions that will result. For that reason the system was sized to accommodate future additions as previously noted and the control was designed to be as flexible as practicable. In general, the groups of heaters are so controlled that any one or all groups may be operated readily either to full capacity or at reduced capacity or may have the operation discontinued, so that the demands of the various seasons of the year and of the ultimate requirements of the system may be met.

Certain safety precautions have been incorporated in the system because of the large number of persons visiting the monument and because of the confining shape of the structure. Extra heavy material has been used for portions of low pressure piping in the interest of safety, long life and structural strength. Insulation has been used on steam piping as a protection against possible injury to adventurous children as well as for the previously noted reasons. A remote manual safety control has been provided for discontinuing the supply of steam to the system in the event of a mechanical failure of any of the piping or heating units.



This blower type unit heater serves the lobby and vestibule at base of monument

In regard to the question of possible excessive updraft due to stack effect with the heating system in operation, it should be noted that the reverse condition exists now. In summer, warm air is drawn into the shaft by reverse convection through the windows at the top, is chilled rapidly by the cold mass of the structure, falls down the shaft and escapes outward with considerable velocity through the entrance doors. Although the effect of the introduction of heat into the shaft cannot be predicted accurately, the immediate effect can be anticipated. In view of the computations, it may be a matter of years before equilibrium will be obtained and an updraft established, at which time the flexibility of the controls should allow for operation of the equipment as will be suitable for the conditions that may then be found to prevail.

[Photos by James M. Osborne]

ABBREVIATIONS FOR ENGINEERING TERMS

The American Standards Association, 29 W. 39th St., New York, N. Y., reports completion of the revision of American Standard Abbreviations for Scientific and Engineering Terms (Z 10.1-1941); price 35c. This standard, providing as it does the shorthand of engineering and scientific language, has a very wide application. It includes such common terms as pounds per square inch (psi), dollar ($), and dozen (doz), as well as kilograms per second (kgps), reactive volt-ampere (var), and British thermal unit (Btu).

The standard has been developed by a group of engineers, editors and scientists representing many national organizations.

Chief difference from the earlier edition of the standard is the inclusion of abbreviations for terms not previously covered, and the shortening of certain previously recommended abbreviations. For example, in the new standard, lb per sq in. of the 1932 edition has been shortened to psi (without a period). Other characteristics of the standard are elimination of spaces between word combinations or letters, and elimination of periods except in cases where the abbreviation spells out a common English word. For example the abbreviation for barometric is bar. (note period), while the abbreviation for calorie is cal (without a period).

Another important change which the committee made early in this work was in abbreviations for units in the temperature scales. The abbreviation deg, and its equivalent the degree sign °, have both been dropped in cases where a specific temperature is referred to. Thus, 212 deg. fahr., or 212 deg F, or 212° F, become simply 212 F.

In all its work the committee has made current usage its guide. Consistency has been admitted when convenient and, when not, refuge has been taken in the conviction that consistency in human beings is unnatural. The letter m in various usages stands for minute, or mile, or meter, or milli, or million, or even thousand; per has had to be spelled out in some cases and used as p in others; hour has had to be hr or even h in order to keep from violating general usage. However, the committee's stand has been that there was no advantage, merely for the sake of consistency, in writing miles per hour, or miles per hr, when mph was so widely accepted.

HEATING, PIPING AND AIR CONDITIONING follows these standard abbreviations in its editorial columns.

ENGINEERING TRAINING FOR DEFENSE

First sessions of new classes giving engineering training for defense were held last month at Northwestern Technological Institute. Organized under the auspices of the United States office of education, classes are given in ventilation and air conditioning; combustion engines; and elements of structural inspection, testing and design.

Ventilation and Air Conditioning

The purpose of the ventilation and air conditioning course is to give an adequate background to handle both the general ventilation problems which arise in industrial plants and when poisonous or objectionable vapors and fire hazards must be handled, and also to develop the methods of air conditioning and control when processing dangerous chemicals (explosives) or precision materials and products.

Complete information on these courses can be obtained by addressing the Defense Training Program, Swift Hall, Northwestern University, Evanston, Ill.

TESTING and DRYING REFRIGERATION SYSTEMS

By Melvin A. Ramsey*

TESTING—A novel method of testing refrigeration and air conditioning systems for tightness, using air pressure, is described by Mr. Ramsey; it was developed for use in testing two 270 ton installations, and since then has been applied successfully as routine procedure on several other jobs. . . . The method of testing is fully explained, as is the method of drying out the system. In discussing the means by which moisture is removed from the system, the author explains the fundamentals involved with a few brief calculations given in the text. . . . Mr. Ramsey concludes by summarizing the advantages of the scheme he outlines. Refrigeration piping engineers and contractors will find this article as useful as will those concerned only with air conditioning. (*Note*—In his manuscript, Mr. Ramsey gave pressures in millimeters of mercury as he has become accustomed to using them in testing South American installations, and as they are convenient because they require fewer zeros after the decimal point than do pressures in inches of mercury. The pressures have been converted, however, to inches of mercury in the text)

TESTING a refrigeration system for tightness and drying it are both of major importance to satisfactory and economical operation of the plant. Many methods of performing these operations are inconvenient and expensive, particularly with the larger installations. In connection with two 270 ton air conditioning installations it was thought wise to study the problem with the hope of simplifying the procedure. The method which was evolved proved to be simple and effective and requires the use of no other gas than air. Its use on other jobs since the first two has become a routine which is easily and almost automatically followed by the mechanic in charge.

To use refrigerant for the test for a tight system is not too unsatisfactory with small installations and sometimes it is helpful to use a little to assist in the location of a small leak which cannot readily be found with soap and water when the system is under pressure. However, to fill a large system not known to be reasonably tight with refrigerant might be decidedly wasteful. Also, the experience in testing the 270 ton installations indicates that a much surer test results from the maintenance for several days of a pressure well above the highest regular operating pressure. Naturally, a test at such pressures would be impossible at normal room temperatures if using refrigerant gas for testing.

The reason for maintaining the test pressure for several days is indicated by the results of tests on the particular installations mentioned. Although at 180 psi (pounds per square inch) the pressure loss was very small during the first two days, the rate of loss increased later. Rechecking the joints with soap and water indicated that at least two or three which did not leak during the first two days started to leak after that. This points to the conclusion that continuous pressure opens up some joints. Undoubtedly it is advisable to find these potential leaks and repair them before charging the system. The test pressure used on this installation was about twice the normal condensing pressure and about 5 or 10 psi below the setting of the safety valves.

In a number of cases carbon dioxide or nitrogen have been recommended instead of air for testing. However, these gases cost more than air and seem too expensive for installations where there are fair sized shell and tube condensers and coolers to be filled. The only apparent advantage in the use of these gases instead of air is that they contain practically no water and thus no water is added to the system when building up the test pressure.

Pumping Air Into System

The use of a compressor on the job to pump room air directly into the system is cheap and advantageous in eliminating the handling of tanks and the chance of running short of gas just when the work is about to be finished, particularly if the shortage occurs at some unusual hour of the night when it can't easily be remedied. (Somehow gas always seems to run low at those annoying times). Although the water added when pumping air directly into the system is not too serious, it is certainly better not to add it because it isn't too good for some parts and there is more to remove when drying the system. The only requirement, then, is to remove enough water from the air before it enters the system to assure that none will condense there. This can be accomplished by compressing the

*Arnott & Co., Ltd.

air in a separate receiver to a pressure above the highest that will be used for the test and then letting it expand into the system from the air receiver. Of course, the air in the receiver must be near room temperature when it leaves.

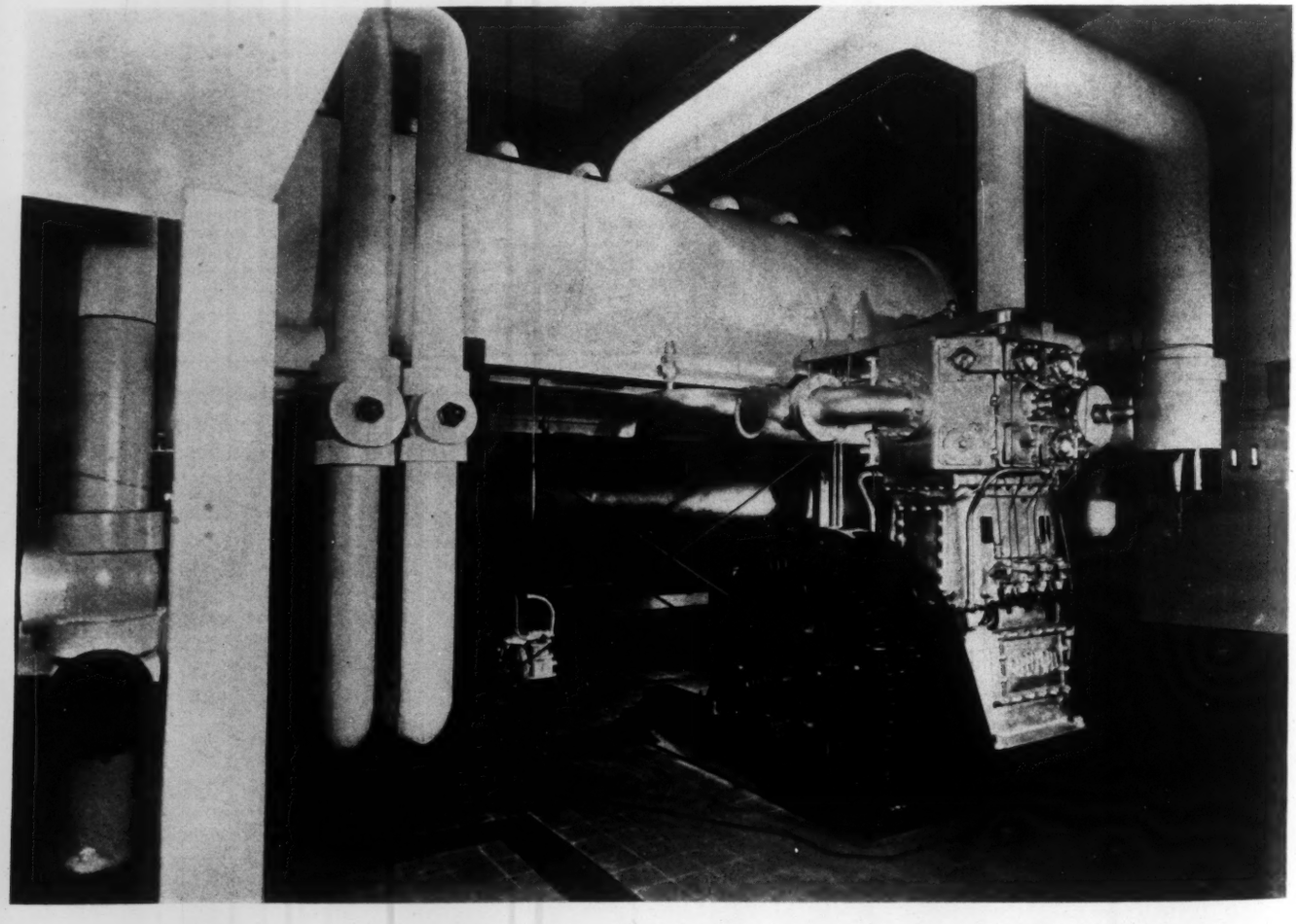
In the case of the 180 psi test, the receiver pressure was built up to a little over 225 psi and then, allowing the compressor to continue, the valve between the receiver and the system was opened to the point where the leak balanced the compressor capacity and kept the receiver pressure between 225 and 250 psi. The air leaves the receiver at its highest point and at the lowest point a valve is provided to drain off the water which accumulates. By this means the moisture in the mixture which enters the refrigeration system is saturated in the receiver at about room temperature. In expanding into the refrigeration system its pressure is reduced to less than that corresponding to saturation at room temperature and no condensation takes place in the refrigeration plant to add to the water to be removed from it.

To assist in locating leaks, gages should be installed, at least temporarily, in each section which can be isolated by valves. As soon as the desired pressure is reached the valves dividing the system should be closed so that time is not wasted making soap and water tests on joints in a section which shows no loss in pressure and therefore does not leak. In one of the 270 ton systems, this sectionalizing was very valuable because there was a leak which could not be found with soap and water. The sectionalizing made it possible to know that this leak was in the compressor section. Removal of the air from this section and the injection of "Freon" made it possible to locate the leak with a halide torch. The leak was in the joint of one of the large cover plates where it was difficult to make a good soap and water seal to find the leak in that manner. After the leak was repaired the compressor was connected to the rest of the system and the pressure raised the slight amount necessary to bring it again to 180 psi. The final pressure loss on both of these particular installations was less than 2 psi from 180 psi over a period of seven days.

The compressor used to inject this air at the pressures indicated was a standard small two cylinder, single stage refrigeration compressor displacing about 100 cfh and driven by a ½ hp motor. Undoubtedly a two stage compressor would be better for pressures as high as 250 psi, but one was not available and the whole power cost is too small to consider any expense for a higher efficiency.

Drying the System

After the system is thoroughly tight and the test pressure released,



View of refrigeration equipment for air conditioning the new building for the ministry of finance in Buenos Aires. This is one of the installations for which the testing and drying method described in the text was developed. A feature of the installation is the use of gas leakage units for maintaining atmospheric pressure in such places as the compressor stuffing box, "Freon" liquid circulating pump stuffing box, etc., to prevent undue refrigerant losses at these points. The evacuating unit is below the condenser

it is ready to be dried. Of course, no system can be perfectly dry but the quantity of moisture must be so small that none exists as a liquid and the vapor pressure of that which does exist should be lower than that corresponding to the saturation pressure at the lowest operating temperatures.

Let us assume now that we have a refrigeration system which is to operate (part of the time at least) with 0 F refrigerant temperature. This means that the vapor pressure must not be more than 0.0449 in. of mercury (Hg). This may seem too low a pressure to obtain on a large system and a pump for a pressure below this might be considered too expensive. Fortunately there is no need for such a low total pressure.

Suppose that the system to be dried is in a room having a temperature of 70 F and therefore has a temperature of 70 F. The vacuum pump available has a capacity sufficient to reduce the pressure in the system to 0.236 in. Hg absolute. The saturation pressure corresponding to 70 F is 0.7392 in. Hg for water.

The vacuum pump is connected and begins removing the mixture of air and water vapor from the system. If there is liquid water in the system little evaporation will take place until the pressure reaches 0.7392 in. Hg, at which point the water will begin to boil.

If, during the reduction in pressure, a time-pressure record is kept it will be noted that, although the rate of fall diminishes slightly, there is a continuous fall, at least until a pressure of 0.7392 in. Hg is reached. At this point, if liquid water exists in the system there will be no further drop in pressure until all the water is evaporated. This assumes, of course, that the rate of evaporation is not so great as to take away heat from the liquid faster than the liquid gets it from its surroundings, thus reducing its temperature. This rate of evaporation is not likely to be the case unless the liquid is in some well insulated point in the system.

The evacuating now continues and finally all the water is evaporated (or it is chilled, reducing its vapor pressure) and the pressure in the system falls to the 0.236 in. Hg limit which has been assumed. The pump should now be stopped and the valve closed to the outlet from the system to find out whether or not the pressure remains constant. If it does over a period of a few hours, there is no leak and no liquid water remains in the system. If the pressure rises to some point well under 0.7392 in. Hg and stops there it indicates that there is no leak, but that some liquid water existed at about 40 F when the 0.236 in. Hg pressure was obtained and that it is now all evaporated. If the pressure rises to 0.7392 in. Hg and stops rising at that point, there is no leak, but water at about 40 F existed and has now risen to 70 F, some or all being evaporated. The only way to find out whether or not it was all, is again to reduce the pressure to 0.236 in. Hg and recheck.

Pressures of Saturated Water Vapor

Values taken from tables by William Goodman published in January, 1938, HPAC

Temperature, F	Pressure, In. Hg
—40	0.005587
—30	0.009824
—20	0.01674
—10	0.02777
0	0.04491
10	0.07096
20	0.1098
30	0.1664
40	0.2478
50	0.3626
60	0.5218
70	0.7392
80	1.0321

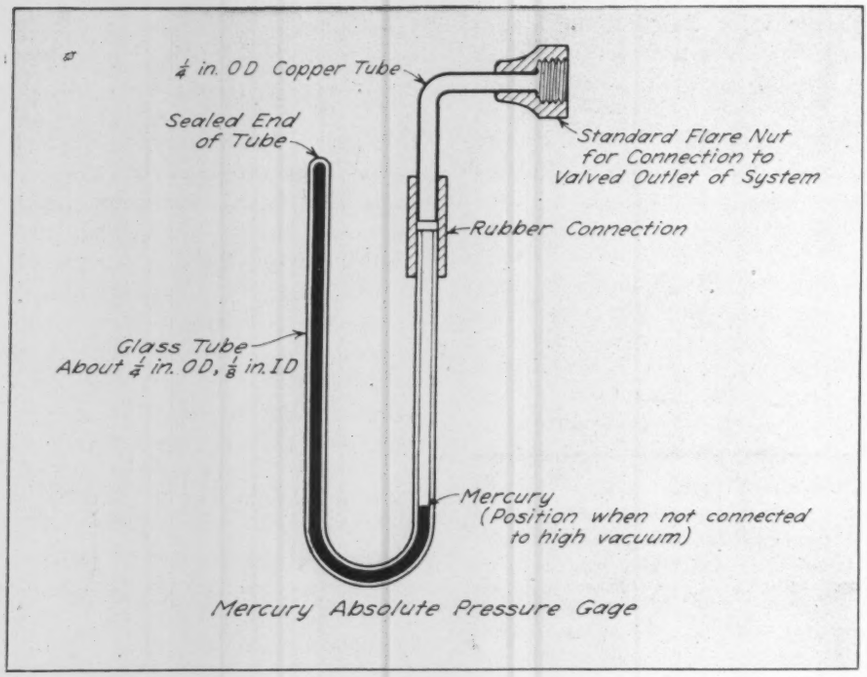
If the pressure rises to well above 0.7392 in. Hg it is a definite indication of a leak, and there is no indication as to whether or not there is liquid water still in the system. If the pressure is again reduced and the rate of reduction continues past 0.7392 in. Hg without change, it is certain that all liquid water has been evaporated and a leak is the only difficulty. A notable change in the rate of reduction at 0.7392 in. Hg of course shows that all water has not been vaporized and continued evacuating and checking are necessary.

When the system holds the 0.236 in. Hg pressure with no more than say 0.15 in. Hg rise in 24 hr, everything is satisfactory and the drying may be completed. (The pressure in one of the large plants mentioned, where delay in completing the building allowed an unusually long time for tests, rose only from 0.236 to 0.590 in. Hg in over six weeks).

The 0.236 in. Hg pressure may be exerted by a mixture of air and water vapor or, if much water existed as a liquid, the water vapor from the boiling water may have displaced most of the air. Since we do not know what part of the gas remaining in the system is water vapor and have no convenient way of determining this, it is wise to assume the worst condition and consider that it is all water vapor.

The water vapor now has a pressure of 0.236 in. Hg and it must be reduced to less than 0.0449 in. Hg. The available pump will not remove any more of the water vapor without assistance. Probably the simplest method of getting this assistance is to open a valve and allow room air slowly to enter the system and mix with the water vapor. Of course, in doing so more water vapor is allowed to enter since the room air may even be saturated. Again it is wise to assume the worst and suppose that the room air has 100 per cent relative humidity at 70 F, which is a vapor pressure of 0.7392 in. Hg. This air should be allowed to enter only until the total pressure in the system rises to one-half an atmosphere, or 15.000 in. Hg. As the mixture expanded to twice its volume at atmospheric pressure the water vapor did also and the vapor pressure in the system has now become $0.7392/2 + 0.236 = 0.606$ in. Hg. The remaining 14.394 in. Hg pressure in the system is due to the air, which, if it entered slowly, should be well mixed with the water vapor to form a reasonably uniform mixture.

The vacuum pump is now started to remove this mixture, or to be more exact, to remove $(15.000 - 0.236) \div 15.000$ or 98.4 per cent of the mixture, since the pressure is reduced only from 15.000 to 0.236 in. Hg. In removing this part of the mixture the pump has not been able to change the proportions of water vapor and air in the mixture. It has therefore removed 98.4 per cent of the air and 98.4 per cent of the water vapor and reduced the total and partial pressures to $100 - 98.4 = 1.6$ per cent of their previous values. The vapor pressure has now become 0.606×1.6 per cent,


¼ in. OD Copper Tube
Sealed End of Tube
Standard Flare Nut for Connection to Valved Outlet of System
Rubber Connection
Glass Tube About ¼ in. OD, ⅛ in. ID
Mercury (Position when not connected to high vacuum)
Mercury Absolute Pressure Gage

or 0.0097 in. Hg which is considerably lower than we require.

It would be possible now again to allow room air to enter the system and, by the same process, reduce the vapor pressure in the system still further, but this reduction is hardly worth the effort. It is doubtful if it is ever necessary to have a system with less than 0.01 in. Hg water vapor pressure because even if the water vapor did all condense under some unusual condition it is doubtful that there would be enough to do any harm.

If however, it is believed necessary to have a still lower vapor pressure there are two simple alternatives, or a combination of the two. The first would be a better vacuum pump. The second would be the injection of air, nitrogen or some other suitable gas, from a tank where the gas has been kept at, say 2000 psi and 70 F.

The pressure of the vapor in this mixture cannot be over 0.7392 in. Hg. When this gas is allowed to enter the system until the total pressure is one atmosphere, or 30.000 in. Hg, the pressure in the system due to the water vapor will be $0.7392 \times (14.696/2000) + 0.0097 = 0.0102$ in. Hg. The vacuum pump can remove $(30.000 - 0.236) \div 30.000 = 99.2$ per cent of this mixture and the pressure due to water vapor after this evacuation will be 0.0102×0.008 in. Hg. At this pressure there would be so little moisture in a system as to be hardly enough to be troublesome even if accumulated all in one place.

Advantages of Method

This method of testing and drying has now been used on several installations with satisfactory results. Its advantages may be summarized as follows:

1) It is not necessary to transport tubes of gas and chemicals and their cost is eliminated. Dryers need not be installed and later removed. Instead a small piston compressor previously mentioned and a small rotary pump are all that need be taken to the job. These units are useful for other operations, such as transferring refrigerant and reducing refrigerant loss when removing air from a charged system, as described in my article in the November, 1938, HPAC, p. 702.

2) The instructions to the installation mechanic are simple. He has merely to make the system tight enough at the pressure the engineer recommends, considering the refrigerant to be used and the condensing temperature. A supervisor can easily check the final pressure and its loss in 24 hr or more. The building up of pressure and time to watch for possible loss can usually take place while other work is being done if the job is properly planned. The mechanic then releases the pressure and obtains a vacuum of, say 0.118 to 0.236 in. Hg absolute pressure which he leaves for a few hours to be sure it holds. Next he allows air to enter slowly until the pressure is a half atmosphere, or 15 in. of vacuum on a standard compound gage. The system is then re-evacuated to say 0.236 in. Hg. and charged.

3) There is a double assurance that the system is tight.

4) There is no dependence placed on the coils, tubing or other materials used being thoroughly dry when purchased and being kept in that condition during storage and installation. Probably the general use of this system would eliminate the need for drying of evaporating coils and refrigerating equipment, which too often are allowed to take up moisture again while being installed. The only thing which seems desirable is to be sure that the coil, tubing or other equipment contains no liquid water as this requires additional time for its removal. Coil drying at the factory could then be accomplished by merely reducing the pressure to well below that corresponding to saturation at the temperature existing. This drying method outlined makes it certain that the moisture has been well removed from every part of the installed system. If charged immediately no water can again enter.

5) The air and other foreign gases as well as the water have been practically completely removed.

Measuring the Pressures

Naturally such low pressures cannot be measured with a standard vacuum gage. An easy way of measuring low absolute pressures is by means of a mercury U gage sealed at one end, such as is illustrated in the sketch on this page.

In evacuating a system in which much water is present, it may be found that the pressure remains constant at that corresponding to the existing temperature for what seems too long a period. This may be due to the accumulation of some water in the oil of the rotary compressor which will tend to spoil the oil seal and also to evaporate and condense with suction and discharge. In case of this difficulty the compressor should be opened, and the oil checked and replaced with fresh oil if necessary.

Heating a Drafting Board 750 Ft Long by 90 Ft Wide

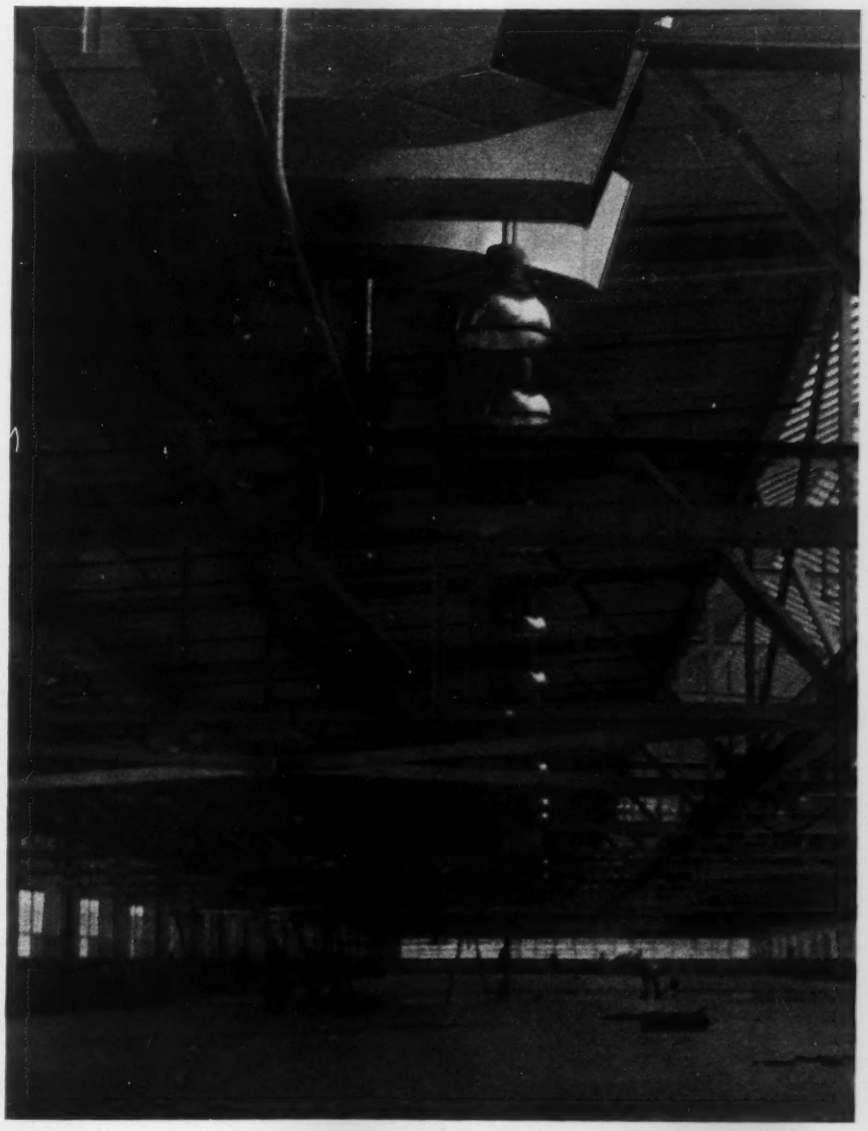
Unit Heaters Solve the Problem of Heating New Mold Loft at Ship Yard . . . . . By C. W. Kimball

A "DRAFTING BOARD" of clear white pine 750 ft long and 90 ft wide is rather unusual, but this arrangement has been completed and forms the floor of the new mold loft of the Bath Iron Works at Hardings Siding, Bath, Maine. As the space below this floor is the plate and angle shop—and rather cold—the problem of heating it so that men could work in comfort was quite interesting.

The mold loft is approximately 20 ft high, including the truss space, and as the walls surrounding it are mostly glass, with considerable glass in the roof, the heat loss had to be figured accordingly. There was also the problem of keeping the room comfortable during the summer months when the sun has a day long sweep at the side walls and roof.

The problem was solved by the installation of seven horizontal centrifugal fan unit heaters up in the trusses, arranged to circulate the air in rotation to insure freedom from drafts and at the same time make sure of heating all parts uniformly. The units are variable speed with preset controllers and the heat delivery is controlled by thermostats. Each unit is rated at 785,000 Btu and 14,680 cfm air flow. This arrangement does away with all pipe coils, radiators or other equipment that would occupy any floor space.

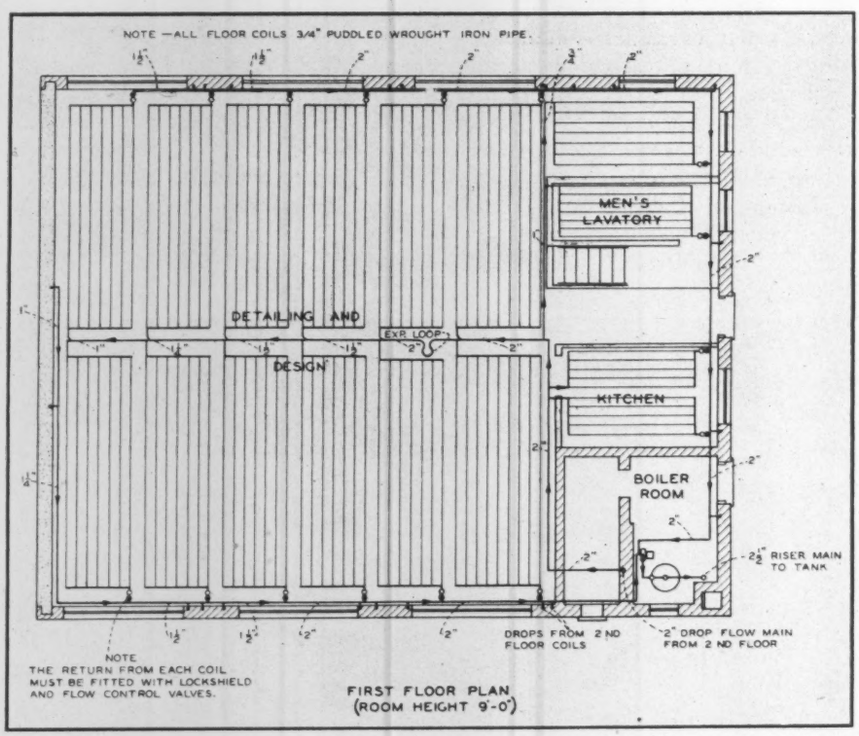
Steam to the units is supplied at 5 to 10 psi from the boiler plant, which is situated in an addition to the main building.



MOLD LOFT—The mold loft of a shipbuilding yard—where the lines of a ship are laid down—is really a huge drafting board and presents an important heating problem, for men must have comfortable conditions for working on the floor. Mr. Kimball (of Richard D. Kimball Co., engineers, and member of HPAC's board of consulting and contributing editors) tells how unit heaters solved the problem without requiring any floor space at one of the nation's most important and busiest yards. . . . In this mold loft, the various frames, plates, braces, etc., are drawn out full size on the floor and then, after all the lines and shapes are trued up and made correct, the lines are transferred to wood approximately ⅜ in. thick. The wood is cut out to these lines, thus making patterns from which the steel for our ships can be shaped

RADIANT HEATING JOB PICTURED

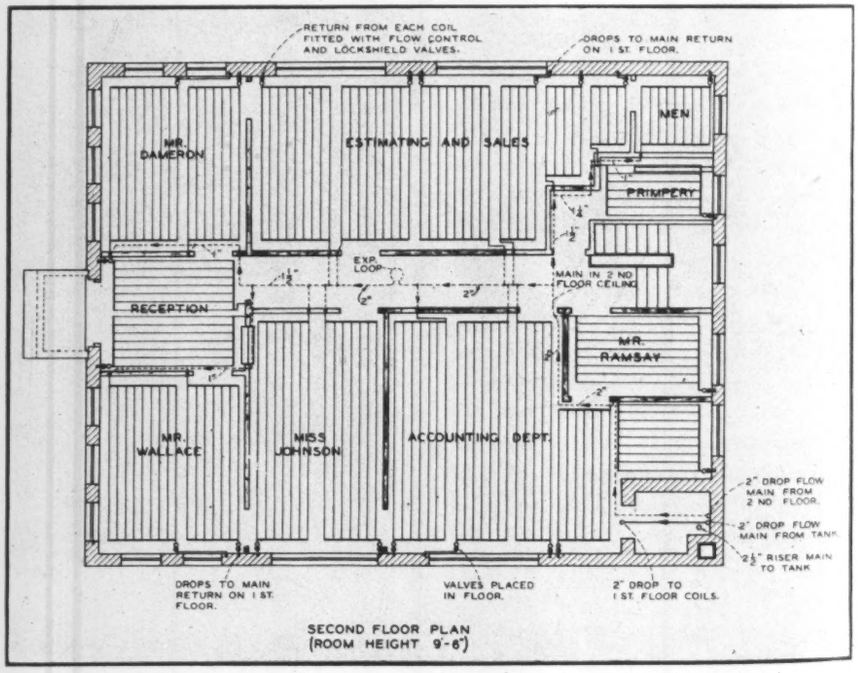
By T. NAPIER ADLAM*

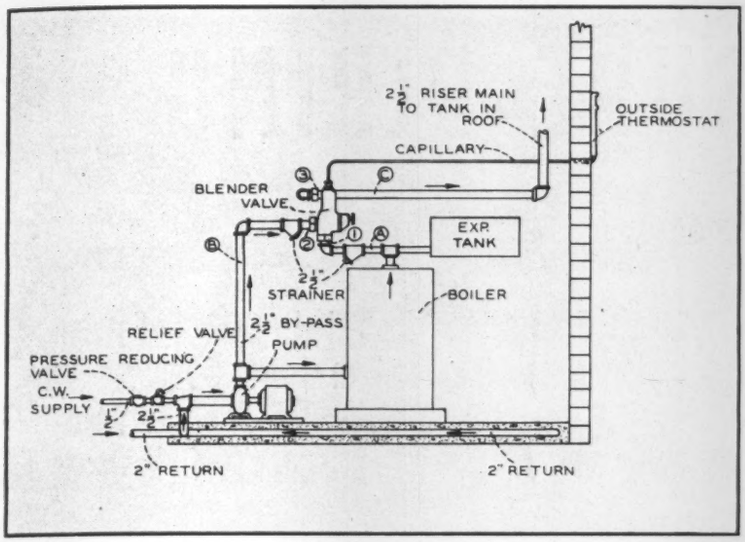

NOTE —ALL FLOOR COILS 3/4" PUDDLED WROUGHT IRON PIPE.
MEN'S LAVATORY
DETAILING AND DESIGN
EXP. LOOP
KITCHEN
BOILER ROOM
2½" RISER MAIN TO TANK
DROPS FROM 2 ND FLOOR COILS
2" DROP FLOW MAIN FROM 2 ND FLOOR
NOTE
THE RETURN FROM EACH COIL MUST BE FITTED WITH LOCKSHIELD AND FLOW CONTROL VALVES.
FIRST FLOOR PLAN
(ROOM HEIGHT 9'-0")

WHEN RADIANT heating comes up for discussion, the installation in the new offices of the Greenville Steel & Foundry Co., Greenville, South Carolina, is usually mentioned. A complete radiant job with pipes embedded in the floors, it was in operation all last winter with satisfactory results. Although complete data aren't available, worthwhile economies in fuel oil consumption over more conventional heating systems are indicated; installation costs compared favorably with other types of heating. . . . The system was designed to give the greatest comfort effect without raising the floor temperature beyond that compatible with the physiological reactions of the most sensitive persons. A letter from the owner shows that the objectives of the design have been achieved. Despite the fact that 20 deg weather was experienced on several occasions, no hot spots in the floor could be detected, and the floor felt cool to the touch although it heated the building. One day, someone opened a wrong valve and 180 deg water was forced through the pipe coils for several hours that night. Even the excessively high temperature caused no apparent cracking of the concrete floor. . . . It is planned to circulate cold water through the floor coils in summer to give a cooling effect

*Chief engineer, Sarco Co., Inc.

The layout of the pipe coils in floor for the first story, and the layout of the pipe coils which are embedded in the floor of the second story, are shown here. The building is a two story structure, about 40 by 50 ft. and is constructed with 12 in. brick walls, the insides of which are painted. The windows are made with steel sashes and single glazing. The ground floor has a 2 in. concrete slab covering on top of 8 in. broken stone fill. The ceiling construction of both floors consists of bar joists, approximately 12 in. deep. The floor of the second story and the roof above it have a 2 in. concrete slab. On the top of the 2 in. concrete slab of the roof a tar and gravel covering forms a composition roof. Insulation is used in the ceiling in order to minimize the transmission of heat


RETURN FROM EACH COIL FITTED WITH FLOW CONTROL AND LOCKSHIELD VALVES.
DROPS TO MAIN RETURN ON 1 ST. FLOOR.
MR. DAMERON
ESTIMATING AND SALES
MEN
PRIMERY
EXP. LOOP
MAIN IN 2 ND FLOOR CEILING
RECEPTION
MR. RAMSAY
MR. WALLACE
MISS JOHNSON
ACCOUNTING DEPT.
2" DROP FLOW MAIN FROM 2 ND FLOOR.
2" DROP FLOW MAIN FROM TANK.
2½" RISER MAIN TO TANK
DROPS TO MAIN RETURN ON 1 ST. FLOOR.
VALVES PLACED IN FLOOR.
2" DROP TO 1 ST. FLOOR COILS.
SECOND FLOOR PLAN
(ROOM HEIGHT 9'-6")



Hot water is used as the heating medium; the circulator is driven by a ¼ hp motor. The system was designed to operate at a maximum flow temperature of 130 F, and the automatic control governs the temperature of the water in accordance with the outside conditions, as follows: Hot water from the boiler passes along pipe *A* and enters the blender at connection *1*, while some of the return water from the circulating pump passes along pipe *B* and enters the blender valve at connection *2*. After the water is blended it passes over a thermostatic bulb in the blender to outlet *3*, where it is conveyed along the pipe *C* to the system. The thermostatic bulb is connected to the outside thermostat, the latter being sensitive to both air temperature and local low temperature radiation, but being screened from direct sun rays. If the combined effect of outside air temperature and low temperature radiation is increased in intensity, this acts on the outside thermostat, which automatically moves the valve to check the flow of hot water from the boiler and simultaneously increase the flow of return water direct from the pump—thereby reducing the temperature of the blended water flowing to the system. If the outside air temperature is reduced, or the radiation from the sun ceases, then the thermostatic bulb allows the valve to move in the opposite direction, thus increasing the flow of hot water from the boiler

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Greenville Steel & Foundry Co. Heats With Floor Coils

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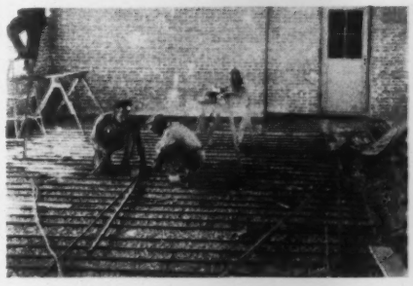
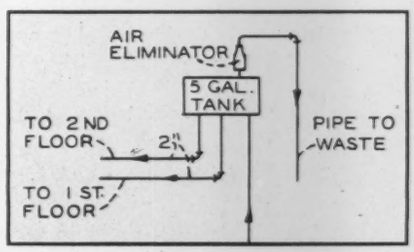
The construction used on the second floor. The prefabricated welded pipe grids were laid over cork insulation while the structural members supporting this floor are steel beams. The pipes were laid with a slight gradient to eliminate the air, and provisions were made for draining the coils when desirable. In designing this system, careful consideration was given to the permissible floor temperature. In order to prevent hot spots the pipes forming the grid coils are ¾ in. throughout. Larger pipes are used for the main circulation to which the various grids are connected. The ¾ in. pipes forming the grids are spaced on 8 in. centers. This gives a perfectly even surface temperature throughout. One of the principal reasons for using small diameter pipes when embedding them in plaster or concrete is that they are less likely to cause cracking if the temperature of the water should go too high



Successive stages of construction of the first floor. Broken stone was first laid on the ground and the pipes carefully laid on the foundation. Additional broken stone was then placed around the pipes, and a concrete slab about 2 in. thick was poured in order to form the finished floor



With such a system it is essential to eliminate all air from the water before it passes to the various pipe coils. To do this efficiently a tank was installed at high level with an automatic air eliminator

HIGH PRESSURE PIPING
at Millers Ford

Steam Piping for 1200 Psi, 900 F Service Installed in 1937 and in 1940 Compared by R. D. Gillespie, Manager of Power Production, and J. D. Williamson, Test Engineer

THE two 30,000 kw, 1200 psi, 900 F topping units installed at the Millers Ford station of the Dayton Power and Light Co.—the first in the summer of 1937 and the second in 1940—offer an opportunity to study the changes that have occurred in high pressure, high temperature steam piping design and installation during a three year period. Most of these changes have to do with the advances in the art of welding carbon molybdenum and plain carbon steel.

Specifications for the steam piping called for materials suitable for continuous operation at 1250 psi ga and 900 F total temperature. In addition to the main piping, this included drips and drains from the main piping up to and including the first valve, connections from flow nozzles and the like.

The 1937 installation required carbon molybdenum seamless steel pipe conforming with ASTM specification A-158-35T, grade P-1. In 1940 the same material was specified, but conforming with the newer ASTM tentative specification A-206-39T, grade P-1, including supplementary requirements S-1 to S-7. The nominal wall thickness in both cases is shown in the accompanying diagrams.

Castings were of carbon molybdenum steel to agree with ASTM tentative specification A-217-39T, grade WCI, with wall thickness not less than the minimum specified for 1500 lb fittings in the American standard for steel pipe flanges and flanged fittings, ASA-B16e-1939.

Details of the various welds are shown in the diagrams. The change in the filler piece and the angle of bevel from 1937 to 1940 was made because the latter required less filler material to be replaced by welding. This resulted in less welding time per weld and less cost, with no sacrifice in the quality of the weld. If this work were being done today, the ends of the pipe would probably not be upset, and a partially recessed 10 deg tapered backing ring would be used, to effect a saving in time and material.

Welding on both installations was made in accordance with the then current requirements of the American tentative standard code for pressure piping, ASA-B31. Only welders who had qualified by passing the tests of this code were used.

Welding was done by the metallic shielded arc process, using direct current, with "reversed polarity" (work negative and electrode positive) for carbon molybdenum steel. "Straight polarity" was used for carbon steel.

After securing proper alignment and spacing, joints were tack welded to insure alignment during the welding operation. The number of tack welds varied from three for pipe up to 3 in. to six for pipe larger than 8 in. The length of the tack welds was approximately twice the thickness of the thinner material joined. These were made with the same electrodes used on the weld and were thoroughly fused to the base metal. Great care was exercised to avoid undercutting. These tacks became part of the final weld so they were carefully inspected and the ends chipped before welding.

Each bead of the weld was chipped, wire brushed, and peened slightly with a blunt tool to relieve residual stress in the weld metal, before the next bead was applied.

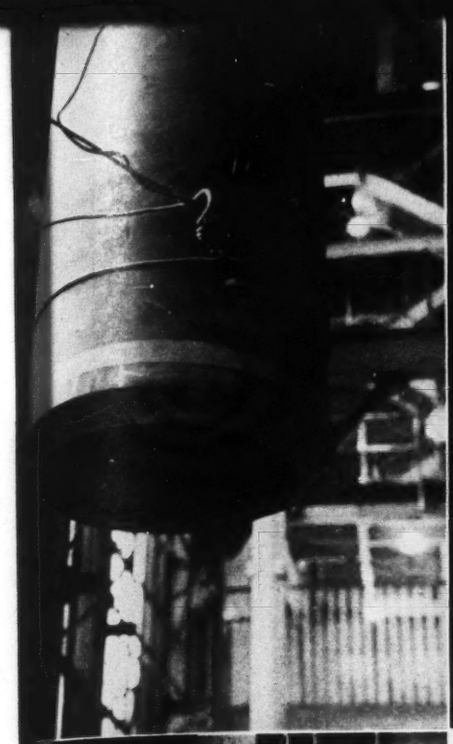
The ends of the pipes and castings were machined to form the contours shown in the diagram. However, flame cut bevels were permitted in the field, provided the work was smooth, true and finished by grinding. Backing rings were not required for shop welds.

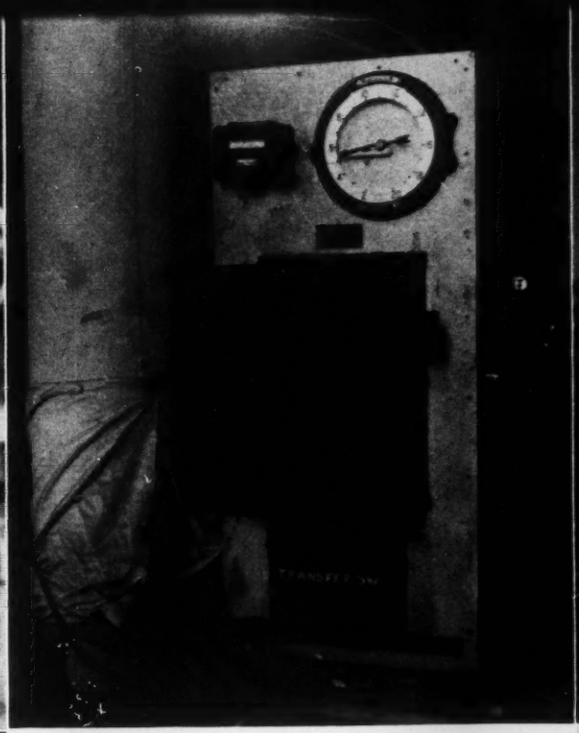
All carbon molybdenum steel welds were preheated to a temperature of 400 F. This consisted of uniformly heating a band equal in width to at least six times the metal thickness on either side of the weld. The temperature was maintained within limits of plus or minus 50 F during the entire welding operation.

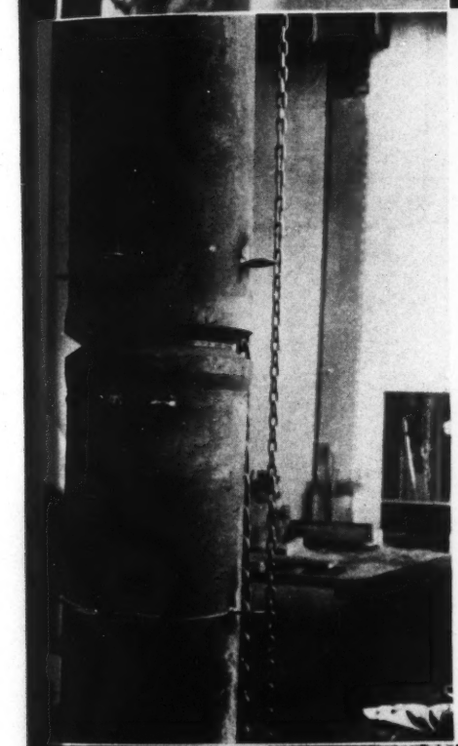
All of these welds were stress relieved by heating the band described above to a temperature of 1150 to 1200 F, and maintaining that temperature two hours per inch of thickness. They were then cooled uniformly and slowly to a temperature of 600 F at a rate not exceeding 200 to 250 F per hour. Below 600 F, cooling in air was permitted.

All preheating and stress relieving was performed by means of electric resistance coils wrapped around the pipe ends. Temperatures and rate of heating and cooling were automatically controlled. A temperature recorder chart was made for each weld showing the temperature prevailing during the entire cycle from the start of preheating to

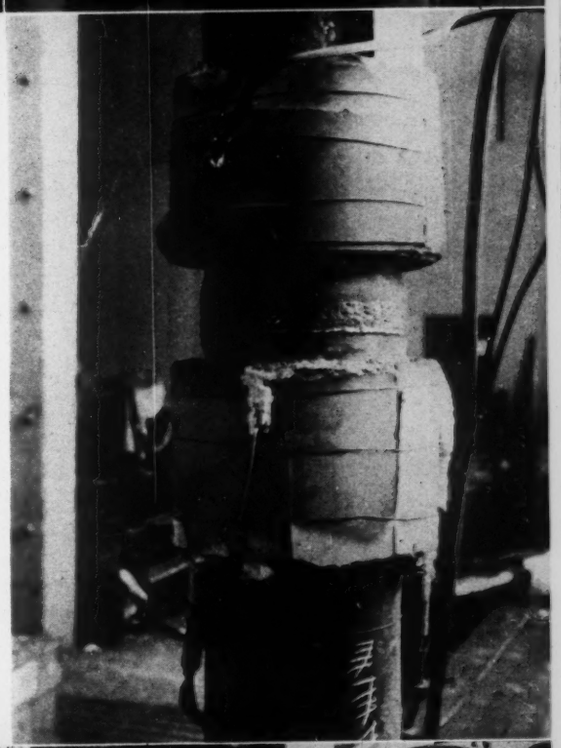
HIGH PRESSURE PIPING—Opportunity is offered by two 30,000 kw, 1200 psi, 900 F topping units installed at the Millers Ford station of the Dayton Power and Light Co. to study changes in high pressure, high temperature steam piping design and installation over the three year period from 1937 to 1940. Most of the changes have to do with advancement in the art of welding carbon moly and plain carbon steel. . . . . The authors describe the specifications, welding details, preheating and stress relief, and valve construction, and present test results on pressure drop in a 12 in. turbine lead. . . . . After almost four years experience with 1200 psi, 900 F steam lines, no difficulties have been encountered due to the welds or other causes, it is stated

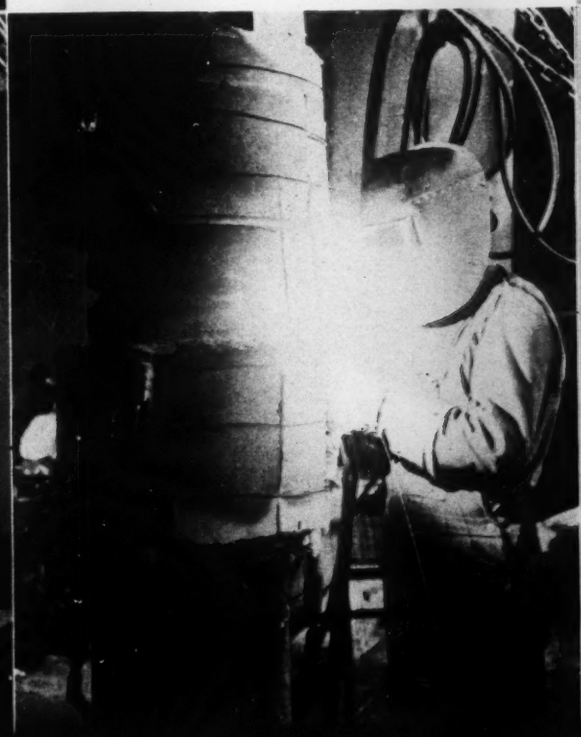


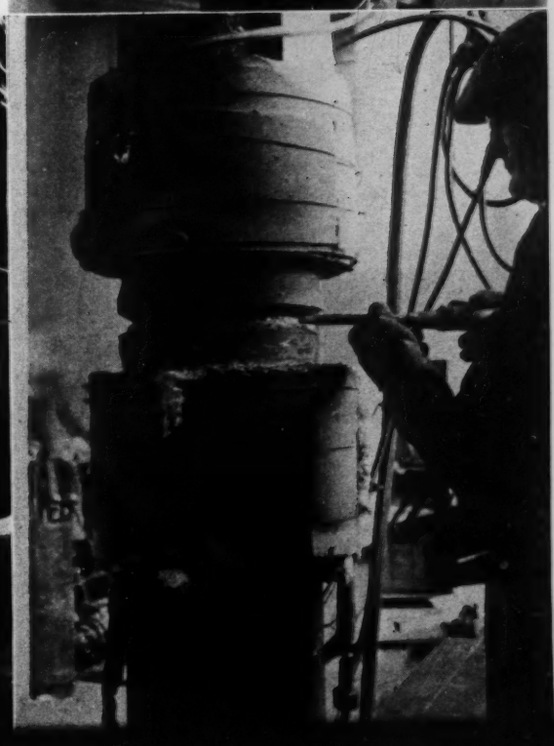




Reading counterclockwise, starting with upper left corner—12 in. carbon moly seamless pipe showing upset end, welding chamfer and recess for chill ring. . . . . Chill ring in place; washers for supporting resistance heating coils; ¼ in. bolt welded on lower pipe for attaching thermocouple. . . . . Heating coils insulated ready to connect to temperature controller for preheating to 400 F continuously throughout the entire welding operation. . . . . Electric arc welding with joint preheated. Thermocouple connection shown at left center. . . . . Chipping and cleaning with joint still preheated. The ninth bead has been completed. . . . . Weld after 38 beads have been completed. One more layer of beads was applied for the required reinforcement beyond upset diameter. . . . . Center portion insulated after completion of welding for use during stress relieving at 1200 F. . . . . Temperature controller for heating coils, showing recorder at right, and heating or cooling rate adjuster







the completion of cooling. Charts were numbered to identify the weld.

Steel fittings were hydrostatic shell tested in the shop at 3500 psi. During the test, blows were struck on each side of the weld and close to it, hard enough to jar the weld but not seriously dent the material. No leaks were found at any weld.

Expansion loops were used on all short runs of this piping. The long runs with risers have no loops, but have long radius ells. Except for the constant support spring hangers used on these lines, all supports, guides and anchors were fabricated on the job.

All gate valves have the body and disc cast from carbon molybdenum alloy steel. The discs have stellite facings on both seat faces and guide ways. The seat rings are forged carbon molybdenum alloy steel with stellite facings on the seat faces.

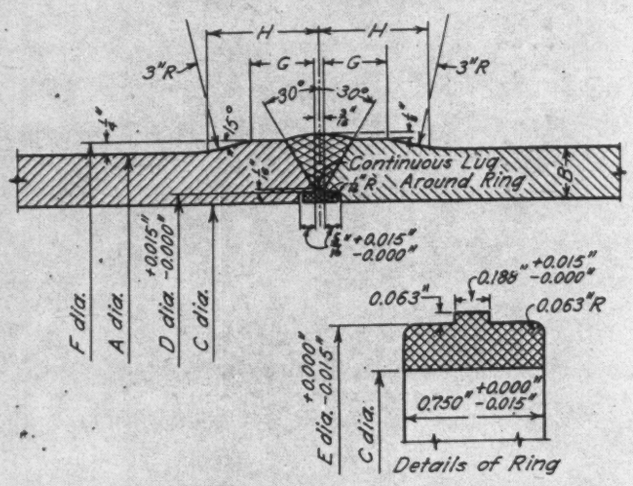
In the 1937 installation, all gate valves 6 in. and larger were equipped with motor drives, and the 1940 installation uses new units of the controlled thrust type made by the same manufacturer. In addition to the valves so equipped in 1937, the 1940 installation also has motor drives on the boiler non-return valves.

In the entire high pressure plant, with the exception of valve bonnets, there are only five bolted flanged joints on 1200 lb, 900 F steam lines. These occur where the 3 in. lines are connected to the throttle valves of the secondary boiler feed pump turbines. A 1/16 in. furniture iron gasket is used in this joint, and it has proved to be entirely satisfactory.

The venturi nozzle at the desuperheating station of the first unit has bolted flanged joints. The nozzle installed last year is welded into the line. A comparison of the two indicates that the latter type of installation is to be preferred.

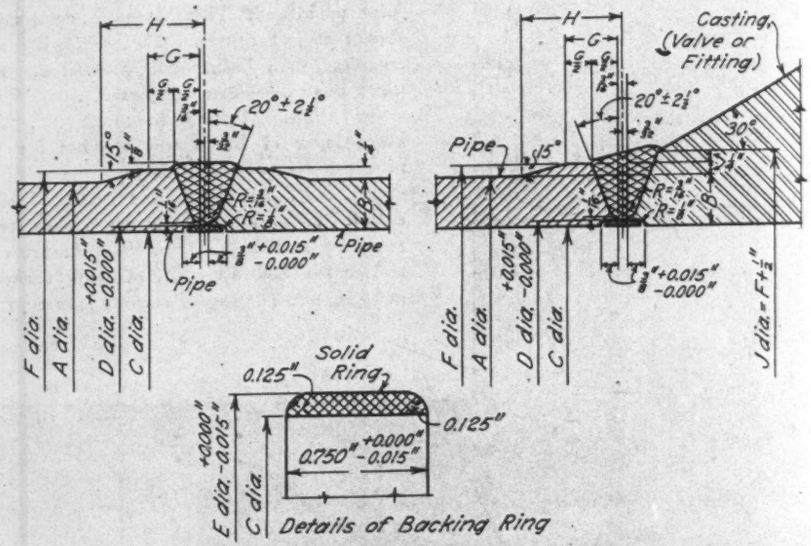
Pressure Drop in Turbine Lead

It is interesting to note the pressure drop in a 12 in. turbine lead (10.126 in. ID). The straight pipe is 143 ft long, and the line has in addition a 12 in. gate valve, two bends of 6 ft radius, two of 5½ ft radius, and two of 5 ft radius. With a flow of 500,000 lb per hr of steam at 900 F, the pressure dropped from 1230 to 1200 psi. These are actual test results.



Pipe Size	Outside Diameter A	Nominal Wall Thickness B	Inside Diameter C	Ring Recess Diameter D	Ring Diameter E	Outside Diameter at Upset F	End of Pipe to 15° Taper G	Minimum Distance to Hanger H
3"	3.500"	0.437"	2.626"	2.876"	2.876"	4.000"	0.563"	1.625"
4"	4.500"	0.531"	3.438"	3.688"	3.688"	5.000"	0.688"	1.750"
6"	6.625"	0.718"	5.189"	5.439"	5.439"	7.125"	0.938"	2.000"
8"	8.625"	0.906"	6.813"	7.063"	7.063"	9.125"	1.125"	2.188"
10"	10.750"	1.125"	8.500"	8.750"	8.750"	11.250"	1.375"	2.438"
12"	12.750"	1.312"	10.126"	10.376"	10.376"	13.250"	1.625"	2.688"

1937 High Pressure Steam Piping



Pipe Size	Outside Diameter A	Nominal Wall Thickness B	Inside Diameter C	Ring Recess Diameter D	Ring Diameter E	Outside Diameter at Upset F	End of Pipe to 15° Taper G	Minimum Distance to Hanger H
3"	3.500"	0.437"	2.626"	2.876"	2.876"	4.000"	0.625"	1.625"
4"	4.500"	0.531"	3.438"	3.688"	3.688"	5.000"	0.688"	1.750"
5"	5.563"	0.625"	4.313"	4.563"	4.563"	6.063"	0.750"	1.813"
6"	6.625"	0.718"	5.189"	5.439"	5.439"	7.125"	0.813"	1.875"
8"	8.625"	0.906"	6.813"	7.063"	7.063"	9.125"	0.938"	2.000"
10"	10.750"	1.125"	8.500"	8.750"	8.750"	11.250"	1.125"	2.125"
12"	12.750"	1.312"	10.126"	10.376"	10.376"	13.250"	1.250"	2.313"

1940 High Pressure Steam Piping

THE WATER-AIR CHART

Its Use for Solving Problems Involving the Exchange of Heat Between Air and Water ..... By William Goodman*

WATER-AIR CHART — The water-air chart provides a graphical method by which a great many types of problems dealing with the exchange of heat between air and water can be greatly clarified and conveniently solved for numerical answers. It can be used not only for problems involving the exchange of heat between air and water in spray chambers and coils, but also for problems dealing with the temperature of the spray water in evaporative condensers and spray coolers, and for problems dealing with the surface temperatures of coils. . . . . The water-air chart provides a particularly clear picture of the action taking place in air washers, cooling towers and coils. With its aid many difficult problems can readily be visualized and solved. This is particularly true of counterflow problems, otherwise difficult to visualize

THE water-air chart shown on p. 222 of the April HPAC provides a graphical method for solving a great many types of problems dealing with the exchange of heat between air and water. The chart itself and its use were explained in April, and a table showing the values of N and θ was presented. In May, the solutions of various types of problems were explained, with illustrative examples, and additional uses of the chart are discussed this month.

Evaporative Condensers

The theory of the evaporative condenser was described in a series of articles in the March, April and May, 1938, issues of HPAC (in which N was defined as GZ/UA). It was shown there that the temperature of the spray water remains constant for any given set of conditions. In order to use the water-air chart (p. 222, April, 1941, HPAC) for finding the temperature of the spray water, the value of N is computed by means of the following equation:

$$N = \frac{UA}{GZ} \quad \text{.....[12]}$$

*Consulting engineer, The Trane Co. Member of HPAC's board of consulting and contributing editors.

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The following example illustrates the use of this equation:

Example 5—Air at an initial wet bulb temperature of 78 F is to be used to condense the refrigerant at a temperature of 96 F. The surface area A of the condenser is 500 sq ft, and the quantity of air G used is 10,000 lb per hr. Assume that $Z = 0.843$, and $U = 19.5$. The barometric pressure is 26 in. Find the temperature of the spray water.

Solution—Referring to Fig. 14, line *1-2* represents the initial enthalpy corresponding to a wet bulb temperature of 78 F and a barometric pressure of 26 in. Vertical line *2-3* represents the condensing temperature of 96 F.

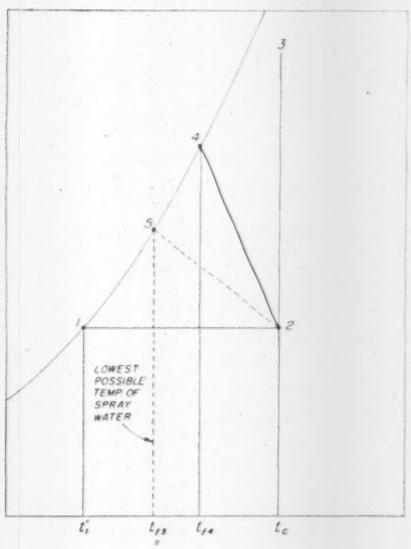
Using Equation 12, $N = (19.5 \times 500) \div (10{,}000 \times 0.843) = 1.16$. From Table 2 (p. 223, April, 1941, HPAC) for $N = 1.16$, $\theta = 61°\text{-}57'$. Draw N-line *2-4* at an angle of 61°-57′ with the horizontal. N-line *2-4* intersects the saturation curve of 26 in. at point *4*. The temperature of the spray water will be 86.5 F as read at point *4*.

The lowest theoretical possible temperature to which the spray water in an evaporative condenser can be cooled is found by drawing an N-line for which the value of N is computed by means of the following equation:

$$N = \frac{cU}{f_s} \quad \text{.....[13]}$$

In actual practice the spray water temperature as determined by means of this equation cannot be obtained, because it requires the circulation of an infinite quantity of air through the condenser. This value of water temperature is of value, however, as an indication of the increase in con-

Fig. 14—Temperature of spray water in evaporative condensers is read at point *4*. Lowest theoretical spray water temperature is read at point *5*



Symbols

A = area of outside surface of condensing coil, sq ft.
B = ratio of external to internal surface of coil.
c = humid specific heat of air ($c = 0.243$ approximately).
c' = specific heat of liquid.
E = efficiency.
f_s = coefficient of heat transfer through air film, Btu per hr per F per sq ft.
f_R = coefficient of heat transfer through liquid film, Btu per hr per F per sq ft.
G = weight of air circulated through condenser, lb per hr.
G' = weight of liquid circulated, lb per lb of dry air.
N = see Table 1, p. 221, April, 1941, HPAC.
t_c = condensing temperature of refrigerant, F.
t_l = temperature of liquid, F.
t_s = surface temperature of coil, F.
t_v = vaporizing temperature of refrigerant, F.
t' = wet bulb temperature of air, F.
θ = angle for N-line.
U = overall coefficient of heat transmission from condensing refrigerant to spray water for an evaporative condenser, Btu per hr per sq ft per F.
Z = see table on p. 167, March, 1938, HPAC.

Numerical subscripts refer to condition of air or liquid at various points in heat transfer process. For example, subscripts 1 and 2 refer respectively to initial and final conditions of air or liquid.

denser capacity that can be obtained with an increase in air quantity.

Example 6—Take the conditions of example 5. Assume $f_s = 12$. Find the theoretical lowest spray water temperature.

Solution—Using equation 13, $N = (0.243 \times 19.5) \div 12 = 0.40$. Referring to Table 2 (p. 223, April, 1941, HPAC) for $N = 0.4$, $\theta = 32°\text{-}54'$. In Fig. 14, draw the N-line *2-5* at an angle of 32°-54′ with the horizontal. Point *5* is located where the N-line intersects the saturation curve for a 26 in. barometer. The theoretically lowest temperature of the spray water is 82.5 F as read at point *5*.

Sprayed Cooling Coils

The sprayed cooling coil is exactly the reverse of the evaporative condenser. In the evaporative condenser, air is heated by a condensing refrigerant; in the sprayed cooler, air is cooled by a vaporizing refrigerant. In both cases, water is constantly recirculated through the apparatus, without being heated or cooled externally. In both cases, the spray water temperature assumes a constant temperature intermediate between the refrigerant temperature and the initial wet bulb temperature of the air. Equations 12 and 13 for the evaporative condenser are also applicable to the sprayed water cooler.

Example 7—Air at an initial wet bulb temperature of 72 F is to be cooled in a sprayed cooler by a refrigerant vaporizing at 42 F. All other values are the same as in example 5. Find the constant temperature that the spray water will assume.

Solution—Referring to Fig. 15, horizontal line *1-2* represents the initial enthalpy corresponding to a wet bulb temperature of 72 F and a barometric pressure of 26 in. Vertical line *2-3* represents the vaporizing temperature of 42 F. Inasmuch as the value of N for this example will be the same as in example 5, θ will also be 61°-57′. Draw N-line *2-4* at an angle of 61°-57′ with the horizontal. N-line *2-4* intersects the saturation curve for 26 in. at point *4*. The temperature of the spray water as read at point *4* will be 54.3 F.

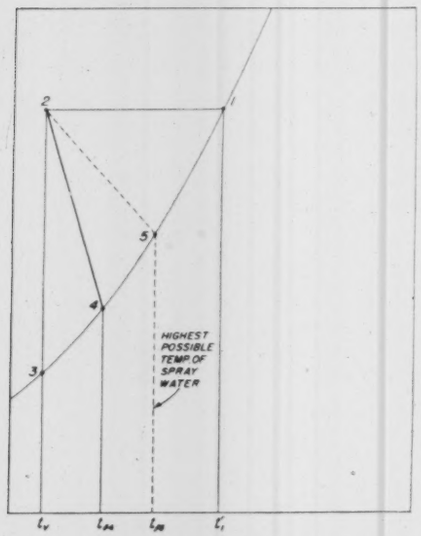


Fig. 15—Temperature of spray water in sprayed cooling coils is read at point 4. Highest theoretical spray water temperature is read at point 5

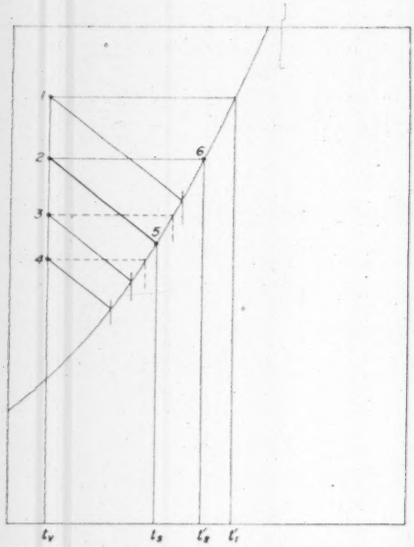


Fig. 16—Method of determining surface temperatures in direct expansion coils

The highest possible temperature that the spray water can theoretically assume can be found by means of equation 13.

Example 8—Take the conditions of examples 5 and 6 and find the highest possible temperature that the spray water can theoretically assume.

Solution—Inasmuch as the value of N is the same as in example 6, $\theta = 32°\text{-}54'$. In Fig. 15, draw N-line *2-5* at an angle of 32°-54′ with the horizontal. Point *5* is located where N-line *2-5* intersects the saturation curve for a 26 in. barometer. The theoretically highest temperature that the spray water will assume is 62.8 F as read at point *5*.

In example 8, the spray water could reach an equilibrium temperature of 62.8 F only if an infinite quantity of air were circulated.

Surface Temperatures of Direct Expansion Cooling Coils

When air is being simultaneously cooled and dehumidified by means of a cold surface, the moisture condensed from the air will form a film on the surface. The temperature of this surface film of moisture is commonly called the *surface temperature* of the coil. The surface temperature of a coil can also be found by means of the water-air chart. As shown in the article on p. 777 of the December, 1938, HPAC an equation may be used to determine the surface temperature at any point in a coil. For this case,

$$N = \frac{cf_R}{Bf_s} \qquad [14]$$

In direct expansion coils, the temperature of the vaporizing refrigerant is assumed to be constant throughout the coil. The surface temperature, on the other hand, varies from point to point on a coil. To determine the surface temperature at any point in a coil, only the vaporizing temperature of the refrigerant and the wet bulb temperature of the air flowing past the point on the coil need be known. Example 9 illustrates the method of determining the surface temperature of a point on a coil.

Example 9—The temperature of the vaporizing refrigerant is 40 F. Assume $f_R = 250$, $f_s = 12$, and $B = 16$. The barometric pressure is 27 in. Find the surface temperature of the coil at the point where the instantaneous wet bulb temperature of the air is 68 F.

Solution—In Fig. 16 locate the horizontal line *2-6* representing the enthalpy corresponding to the instantaneous wet bulb temperature of 68 F. Vertical line *1-4* represents the constant refrigerant temperature of 40 F. Using equation 14, $N = (0.243 \times 250) \div (16 \times 12) = 0.32$. From Table 2 (p. 223, April, 1941, HPAC) for $N = 0.32$, $\theta = 27°\text{-}22'$. Draw the N-line *2-5* at an angle of 27°-22′ with the horizontal. Point *5* is located at the intersection of the N-line with the saturation curve for 27 in. At the point where the wet bulb temperature is 68 F, the surface temperature is 60 F, as read at point *5*.

The wet bulb temperature of the air passing through a coil is constantly decreasing as heat is removed from the air. Inasmuch as the surface temperature of the coil depends upon the wet bulb temperature of the air flowing past that point on the coil, it is apparent that the surface temperature of the coil is constantly decreasing as the wet bulb temperature of the air decreases.

The surface temperature at any point in a coil can be found by drawing a series of parallel N-lines as illustrated in Fig. 16. All of the N-lines originate on the vertical line *1-4* representing the constant va-

porizing temperature. However, points such as *1*, *2*, *3* and *4* are located where the various enthalpies corresponding to the different wet bulb temperatures of the air intersect the vertical line *1-4*. A point such as *1* would correspond to the wet bulb temperature of the air entering the coil, and the intermediate points such as *2*, *3* and *4* would correspond to the wet bulb temperatures of the air at different points in the coil.

Inasmuch as the surfaces of heating coils are dry, the water-air chart cannot be used to determine the surface temperatures of such coils. For dry coils, the surface temperature of the metal forming the coil depends upon the dry bulb temperature of the air—not upon the wet bulb temperature.

Surface Temperatures of Counterflow Cooling Coils

The use of the water-air chart is particularly advantageous in determining the surface temperatures at various points in counterflow cooling coils in which water or other liquids, such as brine, are used to cool and dehumidify air. For this case, the use of the chart will result not only in making the problem simpler, but in considerable saving in time.

For counterflow cooling coils, the value of N is also computed by means of equation 14. However, before the surface temperature at a point can be determined, both the wet bulb temperature of the air and the temperature of the refrigerant at that point must be known. In a counterflow cooling coil, both the wet bulb temperature of the air and the temperature of the refrigerant vary continuously throughout the coil. The temperature of the refrigerant at any point in the coil can be determined either by computation or by means of the method previously described in the section Cooling in Counterflow, p. 293, May, 1941, HPAC. Inasmuch as the temperature of the refrigerant is most easily determined by means of the water-air chart, two lines must be drawn on this chart if the surface temperature in a counterflow cooling coil is to be determined. Thus, in Fig. 17 line *1-2* represents the relationship between the wet bulb temperature of the air and

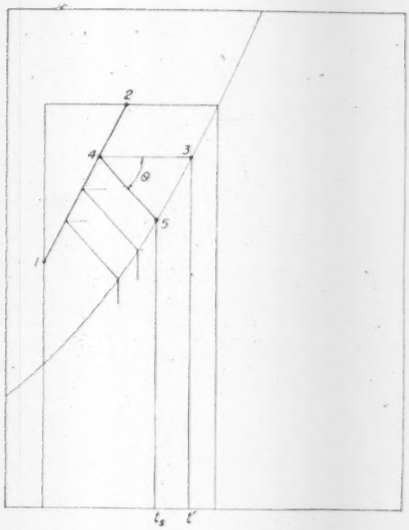


Fig. 17—Method of determining surface temperatures in counterflow coils

the temperature of the refrigerant inside the coil. This line is drawn by the method described in the section Cooling in Counterflow, and illustrated in Fig. 11. The second N-line which is used to determine the surface temperature is drawn in exactly the same manner as described in this month's section on Surface Temperature of Direct Expansion Cooling Coils, except that all of these second N-lines originate on the sloping line *1-2* of Fig. 17 instead of on a vertical line.

The following example illustrates the method of determining the surface temperature at any point on a counterflow cooling coil.

Example 10—Take the conditions of part (*b*) of example 4 on p. 294 of the May, 1941, HPAC. In addition, $f_R = 500$, $f_g = 12$, and $B = 16$. Find the surface temperature of the coil at the point where the instantaneous wet bulb temperature of the air is 64 F.

Solution—The first N-line, *1-2*, of Fig. 17 is drawn in exactly the same manner as described in example 4 and illustrated in Fig. 11. Next locate point *3* where the vertical line of 64 F wet bulb temperature intersects the saturation curve for 28 in. Extend a horizontal enthalpy line through point *3* until it intersects the N-line *1-2* at point *4*. Using equation 14, $N = (0.243 \times 500) \div (16 \times 12) = 0.63$. From Table 2 (p. 223, April, 1941, HPAC) for $N = 0.63$, $\theta = 45°\text{-}32'$. Draw the second N-line *4-5* through point *4* at an angle of 45°-32′ with the horizontal. Point *5* is located where N-line *4-5* intersects the saturation curve for 28 in. The surface temperature is 60.7 F as read at point *5*.

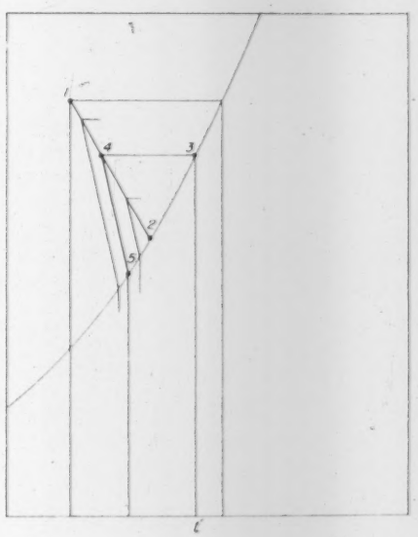
Notice that for problems in counterflow, two N-lines and two values of θ are required. Care should be taken not to confuse these two N-lines. One of the lines shows the relationship between the wet bulb temperature of the air and the temperature of the refrigerant. The second line shows the surface temperature of the coil at any point where the wet bulb temperature of the air and the temperature of the refrigerant are known.

Once the single N-line showing the relationship between the refrigerant temperature and the wet bulb temperature has been drawn, any number of parallel second N-lines—all originating on the one sloping line *1-2*—can then be drawn to determine quickly the surface temperature for a number of different points on the coil. This is very handy when drawing the condition curve for counterflow coils.

Surface Temperatures of Parallel Flow Cooling Coils

To determine the surface temperature in parallel flow cooling coils, two N-lines are also required. First, the N-line representing the relationship between the wet bulb temperature of the air and the temperature of the refrigerant must be drawn. This N-line is drawn in exactly the same manner as described in the section Cooling in Parallel Flow, p. 291, May, 1941, HPAC. The second N-line to be used in determining the surface temperature of the coil is then drawn in the manner described in this month's section on Surface Temperatures of Direct Expansion Cooling Coils.

Fig. 18—Method of determining surface temperatures in parallel flow coils



The value of N is also computed by means of equation 14.

Example 11 illustrates the method of determining the surface temperature at any point in a parallel flow cooling coil.

Example 11—Take the conditions of example 2, p. 291, May, 1941, HPAC. Assume $f_R = 800$, $f_s = 12$, and $B = 16$. Find the surface temperature of the coil at the point where the instantaneous wet bulb temperature is 75 F.

Solution—Referring to Fig. 18, the first N-line, *1-2*, showing the relationship between the wet bulb temperature of the air and the temperature of the refrigerant is drawn in exactly the same manner as described in example 2 and illustrated in Fig. 6. Locate point *3* where the vertical line of 75 F wet bulb temperature crosses the saturation curve for 29 in. Extend the horizontal enthalpy line through point *3* until it intersects N-line *1-2* at point *4*. Using equation 14, $N = (0.243 \times 800) \div (16 \times 12) = 1.01$. From Table 2 (p. 223, April, 1941, HPAC) for $N = 1.01$, $\theta = 58°32'$. Draw the second N-line from point *4* at an angle of 58°-32′ with the horizontal. Point *5* is located where the second N-line *4-5* intersects the saturation curve for 29 in. The surface temperature of the coil is 65.7 F as read at point *5*.

Once the first N-line showing the relationship between the wet bulb temperature of the air and the temperature of the refrigerant has been drawn, a whole series of parallel second N-lines can be drawn for determining the surface temperature at various points throughout the parallel flow coil. All of these parallel second N-lines would originate on the same first N-line *1-2* as illustrated in Fig. 18.

In a counterflow coil, as the air flows through the coil and comes in contact with the refrigerant at progressively lower temperatures, both the wet bulb temperature of the air and the surface temperature of the coil must decrease. In a parallel flow coil, on the other hand, the surface temperature can either increase or decrease as the wet bulb temperature of the air falls.

For example, in Fig. 19 it is apparent that for second N-lines such as *3-4* which lie entirely above the first N-line *1-2*, the surface temperature of the coil will fall as the wet bulb temperature of the air decreases. As point *3* moves down toward point *2*, point *4* representing the surface temperature will also move down along the saturation curve. However, in some cases the

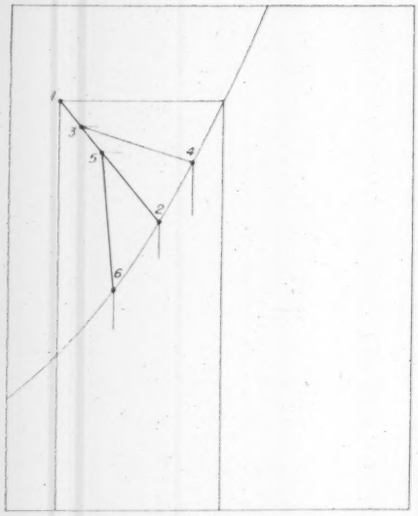


Fig. 19—Surface temperatures in parallel flow coils may either increase or decrease in direction of air flow depending upon relation of first *N*-line *1-2* to second *N*-lines, such as *5-6* or *3-4*

second N-lines will lie below the first N-line *1-2*, as is the case for N-line *5-6* of Fig. 19. For such an N-line, it is obvious that as the wet bulb temperature of the air falls, the surface temperature of the coil will continually increase. As point *5* moves down toward *2*, point *6* representing the surface temperature will move *up* along the saturation curve. This increase in surface temperature is due to the fact that the rise in the temperature of the water more than offsets the drop in the wet bulb temperature of the air.

It is interesting to note that in a parallel flow coil, it is possible to maintain a constant surface temperature throughout the coil; that is, regardless of the changes in the wet bulb temperature of the air and the temperature of the refrigerant, the surface temperature of the coil will be the same throughout the entire length of the coil. If the value of N is exactly the same for both the first and second N-lines, the value of θ will be the same for both lines, and it is evident from Fig. 19 that both N-lines must coincide. In such a case, the surface temperature of the coil will be constant throughout the entire coil. This constant surface temperature of the coil may be read at point *2* of Fig. 19. This point also represents the lowest temperature to which the air can theoretically be cooled.

As previously stated, if the surface temperature of a parallel flow coil is to be constant throughout, the values of N for the first and second N-lines must be equal to each other. Consequently, equating equations 4 and 14

$$G' c' = \frac{cf_R}{Bf_s} \quad \text{.....[15]}$$

When the conditions of equation 15 are satisfied, the surface temperature of a parallel flow coil will be constant throughout.

Determining Film Coefficients

Equation 15 offers interesting possibilities for a method of determining the individual values of f_s and f_R from a test on a parallel flow coil in which moisture is condensed from the air. For this purpose, tests must first be run on a coil with dry surface and the values of U, the overall coefficient of heat transfer, determined. Neglecting the small resistance to the flow of heat offered by the metal, the value of U is represented by the following commonly used equation:

$$\frac{1}{U} = \frac{1}{f_s} + \frac{B}{f_R} \quad \text{.....[16]}$$

If a second test on a parallel flow coil were then run in such a manner that the surface temperature of the coil remained constant, equations 15 and 16 could then be solved simultaneously for the values of f_R and f_s. The problem, therefore, becomes one of so adjusting the flow of air and water through a parallel flow coil that the surface temperature of the coil will be constant. Fortunately, there is a test for determining whether the surface temperature of the coil is constant without the necessity of making difficult surface temperature measurements. If the surface temperature of a coil is constant, the process of cooling and dehumidifying the air will be represented on the psychrometric chart by a straight line (see articles by author in June, 1939—July, 1940, HPAC, and Refrigerating Engineering, October, 1936, p. 225). This fact offers a method of determining when the surface temperature of the coil is constant. The values of the air and water flow during a test must be so adjusted that the points on a psychrometric chart representing the condition of the air as it flows through the coil can be represented by a straight line. This straight line will pass through

[*Continued on p. 363*]

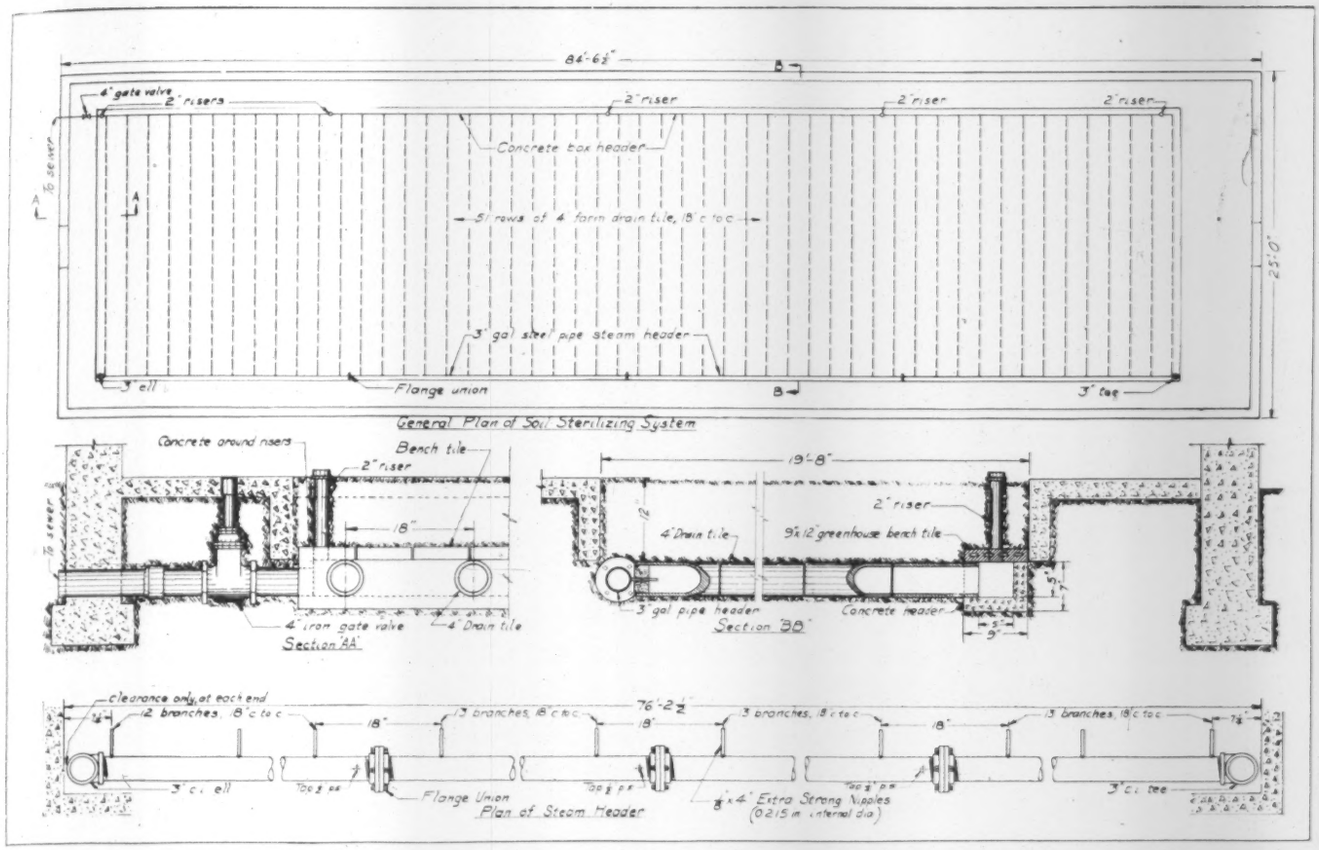


Fig. 1—Plan of buried tile soil sterilizer

SOIL STERILIZING *for Small Greenhouses*

THE plant beds of greenhouses sometimes become infested with small worms which may affect the growth of plants adversely and cause serious losses. Heating the soil with steam is probably the most economical means of destroying these pests.

One of the methods sometimes used for bringing the soil and steam in contact is illustrated in Fig. 1. The system shown was designed for a small greenhouse having a single plant bed 19 ft 8 in. wide and 76 ft 2½ in. long. It consists of approximately 940 ft of 4 in. drain tile laid 12 in. below the surface of the ground and arranged in 51 rows spaced 18 in. center to center.

The rows of tile are supplied with steam through nozzles screwed into a header which runs along one side, the full length of the plant bed. The opposite ends of the rows are connected to a concrete box type header, the chief function of which is to assist in distributing the steam by equalizing the pressure between rows. Steam for both heating the greenhouse and operating the sterilizer is obtained from the underground main of a central heating plant.

Size of Nozzles—In designing the sterilizer, the only problem of any moment was to determine the size of the nozzles so that each row of tile would receive an equal share of all of the steam that could be obtained through the 1½ in. line which connected the greenhouse with the underground main. One-eighth inch extra strong nipples were selected because the internal diameter conformed more closely to the calculated diameter than any other pipe size. Since the nozzles were necessarily small, the extra thickness of the walls was an incidental advantage.

Performance Test—At the time the system was installed, preparations also were made for a performance test. An orifice was inserted for measuring the steam consumption; pressure gages were connected to different parts of the system; and

By L. S. O'Bannon

A "Natural" for the Heating and Piping Engineer and Contractor

a number of thermocouples were prepared for measuring soil temperatures.

Soil Temperatures—Fig. 2 is a plot of the average temperatures observed at different depths (two couples at each depth), while Fig. 3 shows the temperatures observed at the same depth (6 in.) from one side of the plant bed to the other. The distances indicated in the latter chart were measured from the curb on the steam header side of the plant bed.

As indicated in the charts, on the day of the test steam was turned on at 2:10 p. m. and was turned off at 6:50 p. m., thus making the length of the steaming period 4 hr 40 min. The initial temperatures were 15 to 30 F higher than normal, owing to the heat retained in the soil from a preliminary test made seven days earlier.

Surface temperatures were observed with mercury thermometers. In the test shown by the charts, surface temperatures at the time the steam was turned off averaged about 155 F. In the preliminary test, during which the steaming period was 6 hr, surface temperatures averaged about 195 F; while at depths down to the level of the tile, maximum temperatures around 210 F were obtained, such as were observed also in the second test.

In general, if a proper sterilizing temperature be assumed as not less than 130 F, which is in accordance with the recommendations of some authorities, then it may be said that all regions of the plant bed to a depth of a little more than 24 in. had attained this temperature either before the steam was turned off or within a few hours afterward. The lowest curve in Fig. 3 shows that the soil directly over the steam header was slow to heat in comparison with other regions. The chief reason for this was, of course, that there were no joints in the steam header by which the steam could escape directly into the soil. A practical solution for this deficiency would have been to have placed the header in a trench between the foundation and the curb, where the cover of the trench could then be used both as a walkway and as a means of access for inspecting and maintaining the header in good condition.

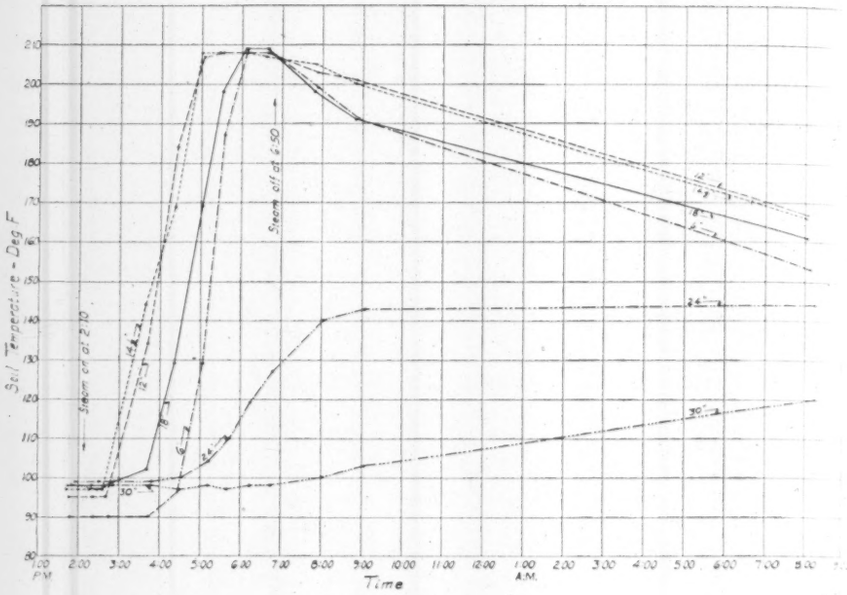
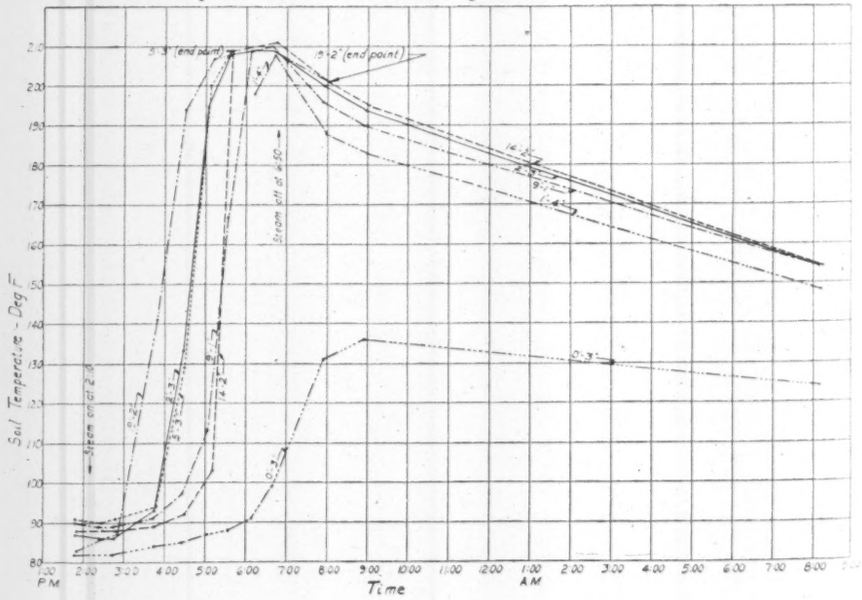


Fig. 2—Soil temperature at different depths plotted against time

Steam Pressures—During the test the pressure within the underground main was practically constant at 50 psi. At the end of the 1½ in. branch to the soil sterilizer the pressure was 14 psi, at the entrance to the header 9 psi, and at the closed end of the header 7 psi.

SOIL STERILIZER — Heating the soil with steam is probably the most economical means of destroying the small worms which sometimes infest the plant beds of greenhouses, affect the growth of plants and cause serious losses. The design and installation of such a sterilizing system falls naturally within the province of heating and piping engineers and contractors, as it represents little more than an extension of the greenhouse heating system. . . . L. S. O'Bannon, research engineer, Kentucky Agricultural Experiment Station, describes a soil sterilizing system here, and gives results of a performance test consisting of steam consumption, pressures in different parts of the system, and soil temperatures at different depths. An important conclusion is that the use of low pressure boilers for greenhouse heating does not preclude the installation of a soil sterilizer of this type, provided the piping is sized so as to distribute the steam properly, maintain a small pressure within the buried tile

Fig. 3—Soil temperature at 6 in. depth at various distances from the steam header side of plant bed. Total width of plant bed between curbs, 19 ft 8 in.



The pressure within the buried tile, as measured by means of a U tube manometer, was about 0.1 or 0.2 in. of water.

Only the pressure within the buried tile is of especial significance, of course; and while it seems somewhat low, it nevertheless was sufficient to cause small geysers of steam to issue from the soil, and (as the graphs show) was sufficient also to do a good job of sterilizing within a reasonable length of time. An important conclusion from these results is, therefore, that the use of low pressure boilers for greenhouse heating, in either old or new installations, does not preclude the installation of a soil sterilizer of this type, provided the piping is sized so as to distribute the steam properly and maintain a small positive pressure within the buried tile.

Steam Consumption—From the orifice readings, the rate of steam consumption was estimated to be 2000 lb per hr. The calculations were made with due regard for modifying factors. Since there is no point in knowing this figure with great precision, it may be taken with confidence as being correct within plus or minus 10 per cent.

Assuming a steaming period of 6 hr, the gross steam consumption amounted to 12,000 lb. Since the area of the plant bed was about 1500 sq ft, 8 lb of steam was used per square foot of plant bed area.

Fig. 4—Three inch steam header, 76 ft long, containing fifty-one ⅛ in. nozzles

As a result of the spading that was necessary when the system was installed, and of the lack of watering, the soil at the beginning of the test was loose and dry. Immediately following the end of the steaming period, samples of earth obtained by digging to the level of the tile were moist enough to ball in the hand, but there was nothing that could be described as mud or mire. Owing to the elevated temperature, the soil dried rapidly.

Acknowledgment—The investigation reported in this paper is in connection with a project of the Kentucky Agricultural Experiment Station and is published by permission of the Director.

References—The following bulletins treat fully the general subject of soil sterilization for greenhouses. The first contains a bibliography listing 37 relevant articles ranging in date from 1889 to 1929:

Control of Root-Knot Nematode in Greenhouses. A. G. Newhall. Bulletin 451, Ohio Agricultural Experiment Station, Wooster, Ohio. (April, 1930).

Soil Treatment for the Control of Diseases in the Greenhouse and Seedbed. A. G. Newhall, Charles Chupp and C. E. F. Guterman. Bulletin 217, College of Agriculture, Cornell University, Ithaca, N. Y. (April, 1931).

The Water-Air Chart

[*Continued from p. 360*]

the point representing the initial condition of the air.

To determine whether a straight line is being obtained on the psychrometric chart for a particular adjustment of the air and water, it would be necessary to split the parallel flow coil into two separate coils in series. The dry and wet bulb temperatures of the air between the two coils could then be measured. Then knowing the initial condition of the air entering the first coil and the final condition of the air leaving the last coil, the three points could be plotted on a psychrometric chart. If the three points fell on a straight line, it would indicate that the coil was then functioning with a constant surface temperature. This would permit equations 15 and 16 to be solved simultaneously for the film coefficients.

This method would have many advantages, since it would completely eliminate the need for assuming the exponents of the air and water velocity or the use of any other empirical factors such as have been needed for all methods proposed heretofore.

Only in running tests on the *wet* parallel surface would it be necessary to adjust the weights of air and water to obtain a straight line on the psychrometre chart. The tests on dry surface used for determining the value of U in equation 16 could be run without any regard to the psychrometric chart other than to make sure that no dehumidification of air was taking place in the coil.

The accuracy of this method hinges on whether the straight line drawn on the psychrometric chart would provide a sufficiently accurate criterion by which to judge whether or not the surface temperature of the coil was constant. This method has not been tried experimentally, but it seems to be worth further investigation to determine its value.

[*This is the third of a series of articles on the water-air chart.*]

C. George Segeler, engineer of utilization, American Gas Association, analyzes some recent studies on temperature and relative humidity conditions in commercial and institutional kitchens, at the hotel, restaurant and commercial sales conference at the Palmer House, Chicago

"Open for Discussion"

Communications to the Editor

AIR CHANGE—ONCE A MAJOR CRITERION

THE EDITOR—

I have read C. M. Ashley's article Air Change—Once a Major Criterion in the March HPAC, and Mr. Frank's and Mr. Stewart's comments last month. All these men realize the importance of the subject of air change, but each sees the answer as it fits his own line.

Whether it be the Black Hole of Calcutta or the most modern theater, the amount of air circulated is of prime importance to the comfort of the occupants. It is true that with the introduction of mechanical means to alter the temperature and humidity within an enclosure the amount of ventilation air is less. It is, I believe, because of this fact that Mr. Ashley says Mr. Frank is confusing the term "air change" with "air change per person." Generally speaking, the air change is less in an enclosure with a light occupancy load; but where a light occupancy occurs, in most cases the volume of the enclosure is small. If you consider an extreme where the volume is very large yet the occupancy is very low, the theoretical air change would be small when considering air change per person only. Yet we know from past experience that this is not the case. The air change must be kept sufficiently high and in many cases above the actual amounts necessary on a per person base to give satisfactory results.

As yet there have been no basic standards presented that attack the problem in a sound practical way. Just where to stop on air change is left to the judgment of the designing engineer. The way he handles the case depends on the type of equipment with which he is familiar.

Mr. Frank, selling ventilation fans, sees the Black Hole of Calcutta handled with large quantities of air, much more than is actually necessary per person but enough to hold down heat gain as much as possible. Mr. Ashley, selling mechanical equipment to alter temperature and humidity, sees the same condition handled with a small quantity of ventilation air; but no doubt he had in mind mechanical equipment to hold down heat gain, although he didn't say as much.

Somewhere between these two extremes lies the answer. An enclosure must not only have sufficient air per person but there must also be enough to keep the air from stagnation. It will stagnate much in the same manner as water in a cistern without an agitator.—WILL D. SAMPSON, Will D. Sampson and Associates, consulting engineers.

LEADING WITH HIS CHIN

THE EDITOR—

C. M. Ashley is leading with his chin in his air change article in the March HPAC! So "air change" is to be "gone with the wind" is it? Well, it seems to me he proves just the opposite in his little dissertation.

I never heard of an engineer designing his air handling volume on the basis of a definite air change, but I do insist that an engineer cross-check the cfm arrived at for heating or cooling load, just to be sure he does not have a monstrosity, such as described by the esteemed author. There is nothing wrong with Fig. 1 of his article except a waste of air handling equipment. If inlet air is desired close to room air, you might use it, but a quick check on air change would say "To hell with 12 or 15 fpm velocity! Let's use less air, say a 3 min (not three per minute) air change."

Likewise, in Fig. 2, Mr. Ashley permits his trained arrows to lead him astray. Will air at 2000 fpm velocity from a nozzle for instance, induce 10 times its own volume? How and where? Look at Fig. 2. If 70 per cent of the cross-section area is used by the return air, that leaves 30 per cent for the supply air and Mr. Ashley would be the first one to say that the air would not follow the arrows in this particular room. If, from a Btu standpoint, only 860 cfm is required for this room, the quick check on air change would prevent the installation of another monument to an engineer who thought that figures couldn't lie.

Seriously, there is a spot for air change and that's for a cross-check. Any cfm arrived at through "pure" science that would give ridiculously high or low overall air change, should be "viewed with alarm." But I'll agree 100 per cent with Mr. Ashley, that you can't start from there.

So far as using air change on outside fresh air volumes is concerned, that's a joke. I've seen outside air short circuited from a toilet or kitchen exhaust hood over to the supply air intake. So when we speak of odors, odor dilution, fresh air, etc., we'd better be careful where we get the fresh air and how fresh it is. Air change means nothing except that it might offset natural infiltration, but what is "natural infiltration"?

Would you like to know my "pet

Readers of HEATING, PIPING AND AIR CONDITIONING are invited to contribute their views to these "Open for Discussion" pages, which will appear from time to time. Comments on articles we have published, expressions of timely interest on developments in design, installation, operation or maintenance of heating, piping or air conditioning systems, other remarks of value to HPAC's readers — all will be welcomed by . . . . THE EDITOR.

peeve"? It's dirty equipment, dirty filters, coils, fan impellers, ducts and grilles. Let's speak frankly about these things. Cleanliness of apparatus has a lot more to do with an odor-free room, than the amount of outside air supplied, whether figured on a minimum cfm per person or on an air change basis. Standard apparatus too often has no simple provision for access for cleaning. In fact, some units are installed in locations where you could not get at the coils to clean them without a major operation.

While we are setting up straw men and real men to knock over, I'd like to set up this one. Every air unit should have access doors to get at filters, heating and cooling coils, eliminators, dampers, fans, and large ducts. Operating instructions should include a cleaning schedule. —LESTER T. AVERY, President, Avery Engineering. Co.

THE BLACK HOLE OF CALCUTTA

THE EDITOR—

In the May HPAC, on the "Open for Discussion" pages, J. M. Frank and C. M. Ashley make reference to the need for air change and the lack of ventilation in the Black Hole of Calcutta.

From what I have heard and read on the subject, I would say that most of the deaths were caused by entirely too high a mean radiant temperature, radiation and re-radiation between the bodies of the various individuals concerned and from the walls of the dungeon proper. Lack of ventilation, to my way of thinking, would be a very small contributing cause to the deaths, for the few individuals who were fortunate enough to have their faces directly up to the one tiny opening also died one by one; others took their places at the opening, so that ventilation certainly wasn't the problem. As to air change, it would seem mechanically impossible to get enough air in such a limited space with such a high heat load to possibly make the conditions better.

I would feel, as just suggested, that an unusually high mean radiant temperature, the source of heat being the individuals themselves, was the principal cause of the tragedy.—CARL F. BOESTER, director of housing research, Purdue University.

VENTILATION FOR ELECTROPLATING TANKS

THE EDITOR—

In the article New Data for Practical Design of Ventilation for Electroplating, by William P. Battista, Theodore Hatch and Leonard Greenburg, published in the February, 1941, HPAC the exponent a for the slope of the curve of Fig. 2 on p. 82 was given as 1.625. This is incorrect; the exponent should be the reciprocal, or 0.61.

Thus, the value of the exponent in the equation $V/(V_o - V) = KX^{-a}$ is —0.61, and in the equation $Q/L = KW^a$ it is +0.61. The exponent is therefore less than the theoretical minimum of 1.0 for a line source of suction, which means that the rate of ventilation per square foot of tank area decreases with increasing tank width. This departure from the theoretical minimum is explained by the flange effect of the horizontal surface below the slot which tends to displace the velocity contours in an outward direction.

A more complete form of the equation for determining the rate of ventilation for electroplating tanks than that given in the last line of the first column on p. 84 of the article is:

$$Q/L = K' V W^{0.61}$$

where Q = total rate of ventilation, cfm;
L = total length of slot, feet;
V = controlling minimum velocity over the tank, fpm;
W = tank width, feet;
K' = constant, the value of which varies with the tank shape and with the shape and location of the exhaust slots. For slots on the two long sides of the tank, K' will be smaller than for one slot only since the velocity over the center of the tank with two slots is the vectorial sum of the velocities established by both slots.

Further experimental work is required to determine the proper values of V and K' for various types of electroplating tanks.—THE AUTHORS.

FOR DEFENSE — DRY BLAST SPEEDS STEEL PRODUCTION

Sloss-Sheffield Steel and Iron Co., Birmingham, Ala., has awarded a contract for an air conditioning system capable of extracting nearly 80 tons of water every day from the air fed into blast furnaces. "Dry blast," the controlled conditioning of blast furnace air, not only saves fuel but also creates uniform operating conditions despite weather variations. With air conditioning there is a definite increase in production, a saving in coke consumption and greater uniformity of the finished pig iron.

The Sloss-Sheffield air control system will condition 90,000 cfm of air for two blast furnaces by means of a single turbo-compressor water cooling unit driven by a 900 hp steam turbine. The air blast will pass through a special dehumidifier before entering the furnaces.

Dry blast is not a new idea in steel circles. In 1904, it was demonstrated at the Isabella plant of the Carnegie Steel Co., and in 1907 and 1908 for Warwick Iron and Steel and Toledo Furnace, respectively. The method is of increased importance today as a defense weapon to speed up steel production without time-consuming plant expansion. Another modern dry blast system will shortly be put in operation for the Republic Steel Corp. at Birmingham. — W. E. BARNUM, York Ice Machinery Corp.

Welding Plays a Vital Rôle in Production for Defense

J. H. Romann* Recalls Piping Experiences During First World War, Cites Importance of Welding Advancements Since That Time

EXPERIENCES during the first world war as chief engineer of a shell forging plant in the Chicago area indicated to me—rather forcefully at times—the importance of plant piping services to efficient production for defense. During the entire war period, the plant produced 8 in. and 9.2 in. shells, first for the English government and later for the United States when we entered the war.

We started out making shells on six 750 ton hydraulic presses, with the hydraulic power being created by six 200 gpm pumps driven by six 200 hp motors. In order to maintain a constant pressure of 1350 psi, an accumulator was installed. The pumps were arranged so as to cut in and out in series whenever there was a pressure change due to the starting and stopping of the individual presses.

The piping connections were such that all the pumps delivered into one header and the presses received from one header. In between these two headers were the accumulators and shock absorbers as well as expansion loops. Each feeder line between header and presses was equipped with a separate shock absorber. Since provisions had to be made also for return lines, intakes, etc., it will be evident to anyone familiar with hydraulic and other piping installations that a great many joints and connections had to be made. Although great care was exercised in the design of the plant layout to avoid friction in the system as much as possible by reducing the number of bends, joints and branches to the absolute minimum, nevertheless the whole presented an elaborate system of piping and fittings.

To prevent damage to the piping by the dropping of tools and equipment on it, and to prevent it from being hit by passing objects, concrete trenches covered with steel plates were provided. Because of the size of the flanges and the number of pipes running parallel in one trench, one can imagine the size of trench required to make it possible for a maintenance man to have easy access to the joints. All of these conditions caused us to locate the pumps and motors close by the presses to keep the supply lines as short as possible.

WELDING—Piping—"industry's biggest transportation system"—is a vital service in the plants of the nation engaged in production for defense. Consequently, welding—an art in its infancy at the time of the first world war—assumes an important rôle in the present emergency. . . . Mr. Romann recounts some of his piping problems as chief engineer of a shell forging plant 25 years ago to indicate one of welding's many applications

Capacity Doubled

When the United States entered the war, the capacity of the plant had to be doubled. Seven additional presses of 750 ton capacity with the additional power drives were purchased and the necessary building expansion was undertaken.

In order to keep the two press batteries in one row, the original power units had to be moved. Since thirteen 200 gal pumps and thirteen 200 hp motors with the necessary control equipment required a great deal of space, a separate building was constructed and the entire power equipment placed under one roof, where it could be kept clean and in the best operating condition. With this move the piping layout became one of the most important parts of the plant. The maintenance of this elaborate piping system was by far the most intricate of all operations and one that required great skill to prevent production interruptions.

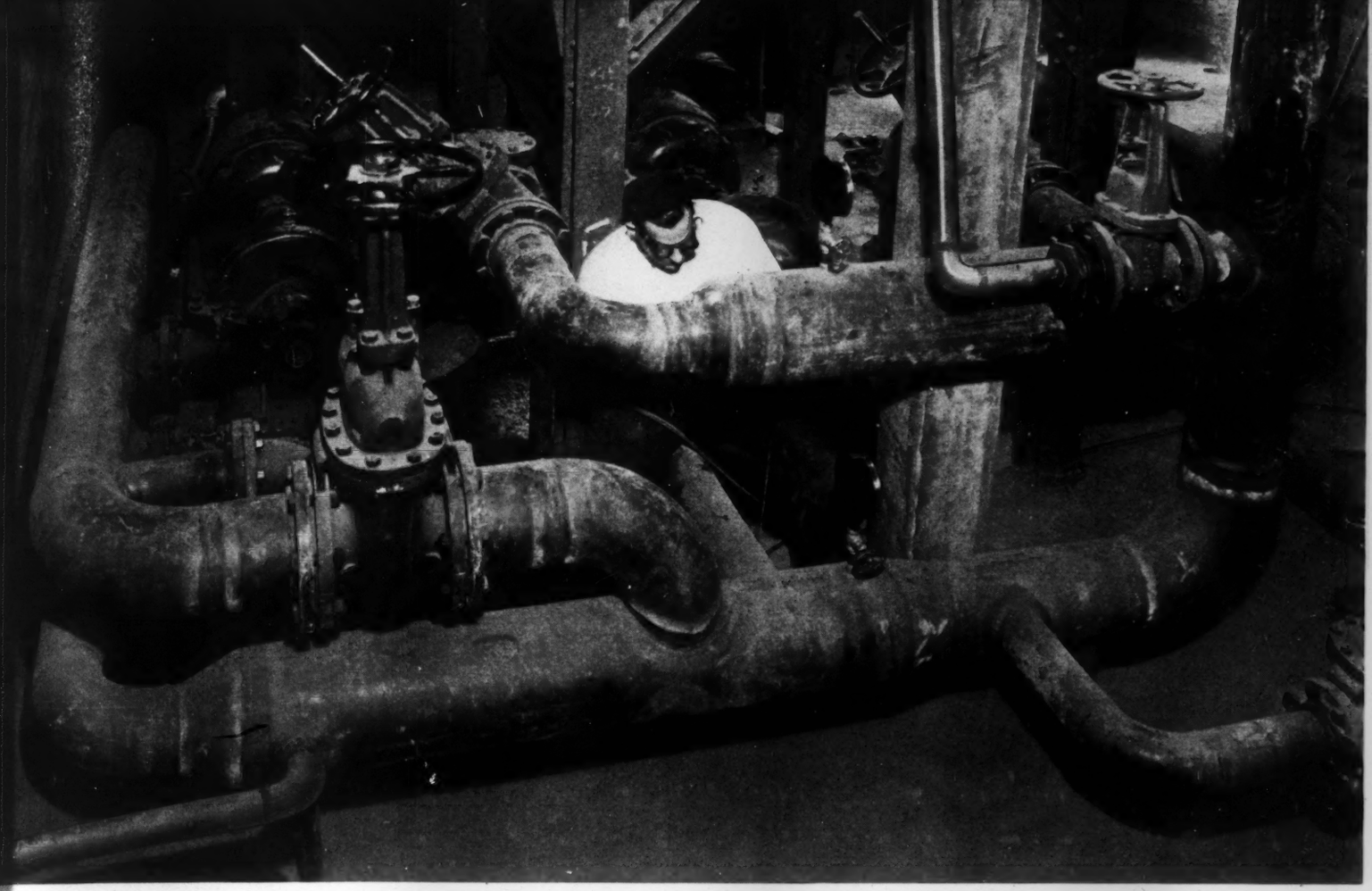
As is well known, one of the causes of leaks in a hydraulic piping system is vibration and shock. When several presses operate simultaneously the consumption of the pressure medium is so great as to cause the accumulator to drop at a fast rate; and then, when the presses come to rest, the movement of the accumulator is arrested suddenly, thereby jarring the entire piping system despite the effectiveness of shock absorbers. Then, too, the opening and closing of the valves causes considerable vibration.

To keep leakage to a minimum, the flanged and bolted couplings were made into male and female halves and were fitted with nonferrous metal rings for packing purposes. However, with all of these and other precautions such as shock absorbers, etc., the operations were often interrupted due to low pressure in the hydraulic system. At the first indication of a substantial pressure drop, such as 200 to 300 psi, an entire battery of presses would have to be shut down, the pipe trenches opened and the piping system carefully inspected over its entire length for small leakage, since two drops per second meant a loss of 150 to 200 psi pressure.

Sunday Maintenance

To keep from losing an appreciable amount of time during the working week, the entire maintenance department would be called out every Sunday, together with a

*The author is head of the experimental laboratory and research department of Tube-Turns, Inc.



A welded piping installation, showing compactness, neatness and adaptability

regular staff of mechanics, to go over this piping system. Every joint would be taken apart, carefully inspected and the necessary repairs or replacements would be made then and there.

Of course the loss of time during the week would mean loss of production. In comparing maintenance labor cost for Sunday work at double time with production loss cost during labor week time, it was found to be much more economical to do the Sunday work even at its exhorbitant labor cost. In working the maintenance department every Sunday as well as on week days, however, it eventually became necessary to employ spell-men to relieve the regulars for a much needed rest.

Welding's Advance

Today, we are fortunate to have advanced in the art of welding. For some years past large industries, such as the oil industry, chemical plants, etc., have adopted welded joints for piping connections. The welding equipment industry, together with the manufacturers of welding fittings and the users of this material, have developed welding practices to a point where the safety factor has reached 100 per cent. This factor, together with the savings in installation and maintenance cost, has made the welding industry one of national importance.

A shell forging plant of today with a hydraulic piping installation does not need to include in the cost the tremendous maintenance expenses caused by leaky joints, since it is now possible to weld practically all lines.

The Marine Field

Recently the naval architects have found the welding of piping in vessels of the greatest importance due to safety and convenience and the tremendous saving in space, which is of much importance in shipbuilding designs, permitting an increase in power and speed. Thus, in the marine field as well as in industry, welding plays an important national defense rôle.

The administration building and engineering office building of the new Thompson Aircraft Products Co. plant, Cleveland, will be air conditioned, using direct expansion "Freon" for cooling; in winter the air conditioning system will be used for tempering the air and heating will be done by hot water convectors or radiators. The cafeteria and dining room systems are designed in the same manner, except that the cooling coils and compressor are being left out for the present. Condensing of the refrigerant is to be by means of evaporative condensers.

The factory will be heated by high pressure steam unit heaters of the down blast type; as the building is designed for black-out operation, tempered air will be introduced through roof units and the air will be exhausted through similar roof ventilators scattered over the full area of the roof, according to the office of Albert Kahn Associated Architects and Engineers.

AIR DISCHARGE from Narrow Slots

Results of Some Field Observations and Experience Summarized by F. F. Stevenson

RECENTLY I was asked to design a slotted air discharge system for industrial air conditioning and found it necessary to get some first-hand information about this method of air distribution. I therefore inspected several installations.

The information given here and in my article in the May HPAC is not the result of laboratory tests, for so far as I was able to find out there are no such data available. It is based on field observations and experience.

Turning Loss

The loss due to the turn of the direction of the air flow and the resulting turbulence in its approach to the slot is difficult to measure and in the latter half of the duct it is not large enough to have an appreciable effect on the design. At the start of the duct, however, where the velocity may range from 1000 to 1500 fpm, this loss is larger and may well be equal to a considerable part of the existing velocity pressure.

The use or non-use of turning vanes or cores has a bearing on this matter, because with the use of turning vanes, the velocity pressure of the air stream can be used (in part, at least) for discharge purposes, while without vanes, none of the velocity pressure is available for discharge. Either way, however, the turning loss need not be considered in respect to the latter half of the duct. The turning loss occurring through the inlet half of the duct will be discussed later.

Shock or "Borda" loss is substantial in expanding sections when air velocities are high or transitions abrupt, but for the work involved here it is negligible.

Static Pressure Regain

In a duct of this sort, static pressure regain due to the decrease in velocity is a matter of interest, especially when the initial velocity is high. The ordinary duct system is frequently designed to effect a regain in static pressure from the fan outlet to the farthest air discharge by sizing the duct so that the friction pressure loss level is decreased at various points along its course. If each transition piece is made in long easy slopes, a considerable regain will be attained. This regain may effect a considerable saving in the fan selection by reducing the total static pressure against which the fan must operate.

With the type of duct we are discussing, the static pressure regain should be considered, in view of the fact that with no turning vanes we use static pressure to provide the energy of discharge.

Static pressure regain due to decrease in velocity is always accompanied by a shock loss, but in the case of a gradually diminishing velocity, such as we encounter here, the loss is too small to be an element in the calculations.

Actual readings show a noticeable increase in the static pressure existing in a duct of this kind between the duct inlet and the "dead end." For instance, consider a duct with a slot 100 ft long, handling 9960 cfm, and with an initial velocity of 1000 fpm. Initial velocity pressure is 0.063 in. water gage and the final velocity pressure is negligible. Neglecting any shock loss as too small for consideration, and computing friction loss by the equation on p. 310 of the May HPAC, assuming an initial friction loss per foot of 0.00037 in, we have a loss of $100 \times 0.00037 \times 0.25 = 0.009$ in.

Then the net static pressure regain is 0.063 in. minus 0.009 in. or 0.054 in. That is to say, whatever the static pressure may be at the beginning of the duct, it will be 0.054 in. greater at the last foot of slot.

Static Pressure and Discharge

The relationship between static pressure and slot discharge pressure is significant. At the far end of duct, with or without turning vanes, we depend entirely upon static pressure to force the air out through the slot. At the fan end of the duct, if turning vanes are used, there is considerable velocity pressure available for discharge purposes, but the static pressure loss caused by the change in direction of flow and turbulence is greater than at the far end, due to the higher air velocity.

Referring again to the duct used in our example above, assume a cross sectional area of approximately 10 sq ft to handle an initial

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NARROW SLOTS — Little information is available on slotted air discharge systems, though this method of air distribution has considerable use in industrial air conditioning work. These jobs are usually designed by rule of thumb, with plenty of provisions for balancing and changing the air distribution after the system is put in operation. . . . . The author was called upon to design a slotted air discharge system and decided to secure first-hand information in the field. He has summarized the results of his "survey" here, and in a previous article published in the May HPAC. Data on required static pressures and on throw of air from slots are included

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Air discharge slots in an air conditioned general office

volume of approximately 10,000 cfm at 1000 fpm velocity.

The face area of the slot, 100 ft in length and assuming its height to be 1 in., is 8.3 sq ft. For the volume of air handled, the face velocity of the air through the slot will therefore be 1200 fpm. Turning vanes are used to direct flow of the air. Assuming a coefficient of discharge of 0.7, we have a discharge velocity of approximately 1700 fpm. The pressure in inches of water needed to produce a velocity of 1700 fpm is 0.181. The friction pressure loss through the edges of the slot is too small to be measured, so that the total pressure needed for discharge purposes is thus 0.181 in. water gage.

At the final section of the slot, the velocity pressure in the duct is negligible, but we have a net static pressure regain available of 0.054 in. Therefore, we must furnish sufficient static pressure at this point to make up the difference between 0.181 in. and 0.054 in., or 0.127 in.

But what static is required at the first section of the slot? We need 0.181 in. for discharge purposes all through the run. However, at the beginning of the slot we have a velocity pressure of 0.063 in. available in the duct. However, we have a static loss, due to the turn in direction and to the turbulent flow, that may be equal to a sizable proportion of the velocity pressure. It may be fair to assume that the turn in direction and turbulence cause a loss of equal to 50 per cent of the velocity pressure, or 0.032 in. Therefore, the total pressure needed at the beginning of the slot run may be taken as 0.181 in. plus 0.032 in. or 0.213 in. We already have 0.063 in. of velocity pressure, so the static pressure needed is 0.213 in. minus 0.063 in. or 0.150 in. water gage.

When discharging air from a streamlined grille or slot of good construction and with the air supply flowing to the grille in the same plane with it (no change in direction of flow) and with the friction loss of the grille negligible, we can assume that the total pressure of air back of the grille (velocity plus static) is wholly converted to discharge velocity pressure. In our example, however, we have the additional static loss due to change in direction and turbulence of flow.

Discharge at Far End

Knowing that it is necessary to have the entering air carry a static pressure of 0.150 in., let us observe how this affects the discharge at the far end.

The net static regain for the run of duct is 0.054 in. and the air enters the duct with a static of 0.150 in. Therefore, at the final run of slot, it will carry a static pressure of 0.150 in. plus 0.054 in. or 0.204 in.

We actually need 0.181 in. for discharge purposes and as both the velocity pressure and turning losses at this point are negligible, we have an excess pressure of 0.204 in. minus 0.181 in. or 0.023 in. This excess is not large. A pressure of 0.204 in. will provide a discharge velocity of approximately

1800 fpm, only 100 fpm greater than required. This may not be critical, but if it is desired to maintain a uniform discharge of air throughout the run, the height of the slot may be made slightly less toward the far end of the duct, increasing the discharge loss, which should balance the system satisfactorily. From the standpoint of conditions at the far end of the duct, we could carry a zero static pressure at the entrance of the duct in many systems of this sort and still have enough pressure at the far end for discharge purposes. But such an arrangement would not be satisfactory for the first section of the duct, as we shall notice later.

If no turning vanes were used in our example, then we could not make use of the air velocity pressure at the duct inlet for discharge purposes, so we should have to carry an entering static pressure of 0.181 in. plus 0.032 in., or 0.213 in. This assumes, for the purposes of the example, that the loss due to the turn and the turbulence is the same whether or not turning vanes are used. This is not strictly true, but the pressures dealt with are low and the variation is not important to the purposes of this discussion.

Turning losses in the latter half of the duct are negligible, so the static pressure available at the end of the run would be 0.054 in. plus 0.213 in. or 0.267 in. This would provide a discharge velocity slightly in excess of 2000 fpm, so we would have recourse to the same remedy as before, *i.e.*, decrease the area of the slot at this point and so increase the discharge pressure requirements.

High Initial Velocity

Trouble is encountered when too high an initial velocity as compared with the static pressure is used in the supply duct. I have a report on one installation with a slot about 150 ft long cut in both sides of an exposed duct and using an initial air velocity of 1600 fpm. The height of the slot was 1⅛ in. and there were no turning vanes. A relatively low discharge velocity was planned and when the system was in operation, readings showed a total pressure of 0.16 in. at the duct inlet, 0.09 in. about halfway along the duct, and 0.07 in. toward the end of the duct. At the entrance and at the halfway mark, the static pressure readings were zero. With this high ratio of velocity to static pressure, trouble was experienced in getting a proper discharge out of the first part of the run of slot. The air tended to flow along through the duct and not to force its way out through the slot.

To make the system operate properly, three things had to be done: (1) the height of the slot was increased at the beginning of the run, reducing discharge pressure requirements there; (2) the height of the slot was decreased toward the end of the run, which increased the discharge pressure requirements there and, incidentally, increased the static pressure that the air must carry when entering the duct; and (3) the fan had to be adjusted to handle the increased static pressure, with no appreciable change in outlet velocity. After the above changes were made, a better air discharge was obtained at the start of the run and a more even balance throughout the whole run of duct was secured.

Slotted systems can usually be balanced in this way and it is best that the slot be designed with some practical method of changing its height or width all along the run.

We must always remember, however, that when the discharge area and the pressure are varied, the throw of the air is affected. Nothing can be done about this, except to effect as good a balance as possible. The greater the discharge velocity and slot area, the greater the throw, but here we are lowering velocity and increasing area, or *vice versa,* which helps to preserve the balance. However, it is desirable to keep initial duct velocity less than the slot discharge velocity for optimum conditions.

Throw from Slot

Calculating the throw of air discharging from a slot forms an interesting phase of this work. Some time ago, I was given a formula credited to Mackey for determining the throw from slots and a couple of tests indicated that it gave fairly good results, though cumbersome to use.

Another formula was given me by H. M. Hendrickson. This is based on the empirical assumption that the throw of air from a narrow slot with a discharge velocity of 1200 fpm will be approximately 20 ft. Starting with this assumption and using a value of 0.8 for the coefficient of discharge, the following equation can be developed: Throw in feet is equal to (cfm) divided by (48 times the square root of the slot face area). The slot face area is taken as a unit value, and in these calculations is 1 sq ft. Space does not permit an exposition of how the equation is developed, but for anyone interested, I will be glad to explain in detail.

The above equation holds up pretty well against actual tests in the field. For slots 2 and 3 in. high, the constant might better be 44 or 40, with 48 serving for slots 1 to 1½ in. high. Of course, for very close engineering calculations, a change in the value of the coefficient of discharge throws a monkey wrench into this equation, but for practical purposes, it seems fairly satisfactory.

As an example of the use of this equation, assume a slot 36 ft in length and 1 in. high delivers 3600 cfm at a face velocity of 1200 fpm. The slot has a face area of 3 sq ft. Using a unit of 1 sq ft of face area, the volume of air delivered from the unit of calculation is 1200 cfm. According to the equation, the throw would be 1200 cfm divided by (48 times the square root of 1), or 25 ft.

Actual values of the air throw taken in the field should be of interest. The readings were taken in a fairly large auditorium with a direct reading air velocity meter. The room was approximately 80 ft x 80 ft, with a 23 ft ceiling. The slot was 2⅞ in. high, with a throat about 8 in. deep, and was about 1½ ft below the ceiling line. The throat had a slight upward spring. Discharge velocity at the slot averaged 1900 fpm.

	Distance from Slot			
	3 ft	18 ft	27 ft	33 ft
Velocity of air stream, fpm.	500	150	75-100	0
Thickness of air stream, ft.	1½	2	3	

I wish to acknowledge with thanks the coöperation of H. M. Hendrickson, F. F. Kravath, A. L. Greenlaw, W. E. Lowell and W. J. Bailey.

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DELTA: *Organized*, 1939. *Headquarters*, New Orleans, La. *President*, C. B. Gamble, 4235 Carrollton Ave. *Secretary*, F. G. Burns, 317 Baronne St.

GOLDEN GATE: *Organized*, 1937. *Headquarters*, San Francisco, Calif. *Meets*, First Tuesday. *President*, N. H. Peterson, 1129 Folsom St. *Secretary*, J. A. Hill, 715 Laurel Ave., Burlingame.

ILLINOIS: *Organized*, 1906. *Headquarters*, Chicago, Ill. *Meets*, Second Monday. *President*, Irving E. Brooke, 189 W. Madison St. *Secretary*, M. W. Bishop, 228 N. La Salle St., Chicago.

IOWA: *Organized*, 1940. *Headquarters*, Des Moines, Ia. *Meets*, Second Tuesday. *President*, Perry LaRue, 629-3rd St. *Secretary*, C. W. Helstrom, 1614 Thompson Ave.

KANSAS CITY: *Organized*, 1917. *Headquarters*, Kansas City, Mo. *Meets*, Second Monday. *President*, W. L. Cassell, 912 Baltimore Ave. *Secretary*, E. M. Jolley, 3001 Fairfax Rd., Kansas City, Kans.

MANITOBA: *Organized*, 1935. *Headquarters*, Winnipeg, Man. *Meets*, Third Thursday. *President*, R. L. Kent, 365 Hargrave St. *Secretary*, F. L. Chester, 179 Bannatyne Ave.

MASSACHUSETTS: *Organized*, 1912. *Headquarters*, Boston, Mass. *Meets*, Third Tuesday. *President*, J. W. Brinton, 1003 Statler Bldg. *Secretary*, E. G. Carrier, 68 High St.

MICHIGAN: *Organized*, 1916. *Headquarters*, Detroit, Mich. *Meets*, First Monday after 10th of Month. *President*, J. S. Kilner, 1091 Seminole Ave. *Secretary*, W. H. Old, 1761 Forest Ave., W.

MINNESOTA: *Organized*, 1918. *Headquarters*, Minneapolis, Minn. *Meets*, First Monday. *President*, M. H. Bjerken, 533 S. 7th St. *Secretary*, D. B. Anderson, 1981 First National Bank Bldg., St. Paul, Minn.

MONTREAL: *Organized*, 1936. *Headquarters*, Montreal, Que. *Meets*, Third Monday. *President*, W. U. Hughes, 1411 Crescent St. *Secretary*, A. M. Peart, 637 Craig St. W.

NEBRASKA: *Organized*, 1940. *Meets*, Second Tuesday. *President*, Frederick Organ, 3724 Mason St., Omaha. *Secretary*, C. E. Wiser, 378 Saunders-Kennedy Bldg., Omaha.

NEW YORK: *Organized*, 1911. *Headquarters*, New York, N. Y. *Meets*, Third Monday in month. *President*, C. S. Pabst, 55 W. 42nd St. *Secretary*, T. W. Reynolds, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

NORTH CAROLINA: *Organized*, 1939. *Headquarters*, Durham, N. C. *President*, W. M. Wallace, II, 111 N. Corcoran St. *Secretary*, F. J. Reed, 263 College Station.

NORTH TEXAS: *Organized*, 1938. *Headquarters*, Dallas, Tex. *Meets*, Second Monday. *President*, R. K. Werner, 316 W. T. Waggoner Bldg., Fort Worth. *Secretary*, M. L. Brown, 3461 Potomac St., Dallas.

NORTHERN OHIO: *Organized*, 1916. *Headquarters*, Cleveland, O. *Meets*, Second Monday. *President*, C. A. McKeeman, Case School of Applied Science. *Secretary*, C. M. H. Kaercher, 3030 Euclid Ave.

OKLAHOMA: *Organized*, 1935. *Headquarters*, Oklahoma City, Okla. *Meets*, Second Monday. *President*, S. L. Rolland, 321 N. Harvey Ave. *Secretary*, A. R. Morin, 2115 Sherman.

ONTARIO: *Organized*, 1922. *Headquarters*, Toronto, Ont. *Meets*, First Monday. *President*, C. Tasker, 43 Queens Park. *Secretary*, H. R. Roth, 57 Bloor St., W.

OREGON: *Organized*, 1939. *Headquarters*, Portland, Ore. *Meets*, Thursday after First Tuesday. *President*, T. E. Taylor, 307 Postal Bldg. *Secretary*, B. W. Moore, 7827 S. E. 35th Ave.

PACIFIC NORTHWEST: *Organized*, 1928. *Headquarters*, Seattle, Wash. *Meets*, Second Tuesday. *President*, M. J. Hauan, 3412-16th South. *Secretary*, H. T. Griffith, 1411 4th Ave. Bldg.

PHILADELPHIA: *Organized*, 1916. *Headquarters*, Philadelphia, Pa. *Meets*, Second Thursday. *President*, H. B. Hedges, State Rd. and Rhawn Sts. *Secretary*, R. D. Touton, 9th and Columbia Ave.

PITTSBURGH: *Organized*, 1919. *Headquarters*, Pittsburgh, Pa. *Meets*, Second Monday. *President*, E. C. Smyers, 148 Jamaica Ave. *Secretary*, T. F. Rockwell, Carnegie Institute of Technology.

ST. LOUIS: *Organized*, 1918. *Headquarters*, St. Louis, Mo. *Meets*, First Tuesday. *President*, C. E. Hartwein, 231 W. Lockwood Ave., Webster Groves, Mo. *Secretary*, C. F. Boester, 101 E. Essex, Kirkwood, Mo.

SOUTH TEXAS: *Organized*, 1938. *Headquarters*, Houston, Texas. *Meets*, Third Friday. *President*, D. S. Cooper, 216 E. Cowan Dr. *Secretary*, A. M. Chase, Jr., Box 359.

SOUTHERN CALIFORNIA: *Organized*, 1930. *Headquarters*, Los Angeles, Calif. *Meets*, Second Wednesday. *President*, H. H. Douglas, 601 W. 5th St. *Secretary*, H. H. Bullock, 212 N. Vignes St.

WASHINGTON, D. C.: *Organized*, 1935. *Headquarters*, Washington, D. C. *Meets*, Second Wednesday. *President*, F. A. Leser, 608 Mills Bldg. *Secretary*, F. M. Thuney, 3707 Georgia Ave., N.W.

WESTERN MICHIGAN: *Organized*, 1931. *Headquarters*, Grand Rapids, Mich. *Meets*, Second Monday. *President*, G. W. Schlichting, 1417 Lovell St., Kalamazoo. *Secretary*, H. J. Metzger, 137 E. Water St., Kalamazoo.

WESTERN NEW YORK: *Organized*, 1919. *Headquarters*, Buffalo, N. Y. *Meets*, Second Monday. *President*, W. R. Heath, 119 Wingate Ave. *Secretary*, Herman Seelbach, Jr., 120 Mariner St.

WISCONSIN: *Organized*, 1922. *Headquarters*, Milwaukee, Wis. *Meets*, Third Monday. *President*, A. S. Krenz, 5114 W. Center St. *Secretary*, H. W. Schreiber, 507 E. Michigan St.

Concerning Conservation of Underground Water with Suggestions for Control

By Noel E. Porter,* Sacramento, Calif.

SUMMARY—Increased usage of air conditioning presents problems in the use and waste of cooling water to municipalities. Commercial areas demand protection of underground water supply by means of regulation. The water rate for cooling equipment can be proportioned to the load resulting in a substantial saving. Ordinances may be provided limiting the rate of disposal to sewers. Regulations governing the use of disposal wells may be required in the future

Introduction

THE rapid increase in comfort air conditioning installations presents definite problems to municipalities concerning water supply and disposal. In many large metropolitan areas legislation has been enacted to limit the use and waste of cooling water.

This paper concerns particularly the upper central and Northern California valley area where underground water is essential to farming, industry, and domestic supply. Well water temperatures and climatic conditions are such that the economic balance generally favors the performing of a large percentage of cooling requirements directly with water through extended surface coils. With the growth of air conditioning comes an ever increasing demand for cooling water—and the possibility of drawing unlimited quantities from innumerable wells in a concentrated area could effect considerable hazard. It follows that methods for regulation are necessary to the protection and advantage of all concerned.

Well Water and Disposal Facilities

Average well water temperatures in the central valley area centering around Modesto and Stockton are 66-68 F for wells from 90 to 300 ft deep. In the Northern valley from Sacramento, North to Chico, water from the 120-140 ft strata is usually 63 F with extreme cases as low as 59 F. Deeper wells yield warmer water, the rise in temperature being from 1 to 2 F per 100 ft of depth below the 140 ft strata.

Test results of a number of wells in Stockton and Sacramento reveal a yield of 10 gpm, or more, per foot of drawdown for an 8 in. diameter well. Water stands at 20 to 30 ft below ground level. The table is subject to a drop in late summer, the amount depending upon seasonal rainfall and snowfall, extent of irrigation from wells, etc.

Disposal facilities in the major localities are listed in Table 1. Sanitary systems require pumping at the sewer farms and are usually limited in capacity for cooling waste. Moreover, clear water produces leaks in the joints of certain types of soil pipe. Storm sewer systems drain by gravity and generally have ample capacity for cooling waste needs. The storm sewer systems are limited in coverage however, and generally serve only the metropolitan districts. In the smaller localities, storm sewers are often routed in alternate streets.

Table 1

City	Disposal Facilities
Modesto	Dual System[a]
Stockton	Dual System
Lodi	Dual System[b]
Sacramento	Single System[c]
Marysville	Dual System[d]
Chico	Dual System[e]

[a]Separate sanitary and storm sewer systems.
[b]Storm sewer system somewhat limited in coverage for metropolitan area.
[c]Single system for both sanitary and storm disposal.
[d]Storm sewer mains located in alleys.
[e]Storm sewer mains routed in alternate streets.
In general, it is assumed that storm sewer systems have ample capacity for cooling waste needs.

To minimize the cost of pumping, nearly every municipality has placed restrictions on cooling waste to the sanitary system, normally a maximum of 5 gpm for each establishment. Where storm sewer systems are economically inaccessible, rates up to 15 gpm are sometimes permitted to be discharged to the gutter. The city of Sacramento, largest in the area, is served by a single sewer system, all of which must be pumped to the river. In December of 1937, Sacramento passed an ordinance limiting waste of cooling water to 1 gpm per foot of frontage for each establishment. This ordinance is effectively a conservation ordinance as it definitely limits the amount of water taken from the ground, and not returned, for each establishment. Any water used over the front foot allowance must be disposed to a return well.

Since inception of the Sacramento ordinance, several return well installations have been made. Definite data have not yet been compiled but the results, so far, have been generally satisfactory. Water is usually supplied from the 120-140 ft strata and returned to the 65-80 ft strata. Some return wells have been drilled to the depth of the supply strata (120-140 ft) and the possibility of mixing the warmer water with the supply may be noticeable in the near future. Return wells have proven unsatisfactory in the lower valley. In the city of Fresno, where many disposal wells have been in operation for some time, there results a very serious rise in the underground water temperature over the summer period.

It is evident that the over use of wells for cooling purposes will in some manner affect the underground

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supply. There is the possibility of effecting a rise in the water temperature, creating sufficient suction to admit brackish water resulting with pollution, or lowering the water table. The latter would produce serious economic difficulties and might cause considerable damage to vegetation and trees that are dependent upon the water table for irrigation. Any of these possible results would seriously hazard all users of underground water.

Effect of Well Water and Climate on Design

Maximum design outdoor temperature for the area is usually taken as 100 deg. The simultaneous maximum design wet-bulb temperature varies gradually from 72 F in the Fresno area down to 67 F in Redding at the northern end of the valley. A typical mean temperature curve for the area is shown in Fig. 1. The curve is plotted using the average of the mean temperatures for the months of June, July, and August, 1940, recorded at six representative hours of the day and as published by the Sacramento Weather Bureau.

The economically available cool water and prevailing wet-bulb temperatures have developed a trend in commercial air conditioning to systems employing large percentages to 100 per cent ventilation air. Internal latent heat load can often be removed simply by providing sufficient ventilation air. Conditioning apparatus is then required to do sensible cooling only, and water cooling may be used to the utmost economic advantage.

Fresh air of 100 per cent is desirable in hotels and office buildings to minimize the transfer of odors and air-borne bacteria. Often, the additional cost for coils and pumping equipment for *all fresh air* is economically advisable when the cost of an adequate return air system is considered. In cities that are principally agricultural centers and free from industrial soot, the filter maintenance problem is reduced to a minimum when conditioners are located on the roofs of buildings. The sales advantages of *all fresh air* are evident.

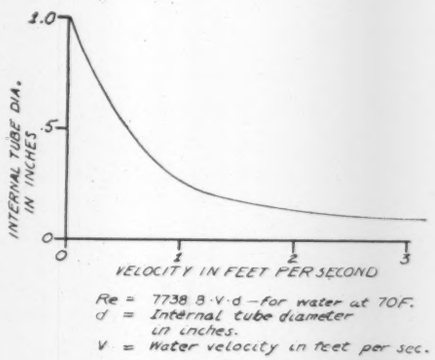

Fig. 2—Plot of internal tube diameter and water velocity for which Reynolds' Number is 2000

Concordant with the foregoing is the fact that the cooling load for the conditioning apparatus varies in a direct relation with the indoor and outdoor temperature difference, or the outdoor temperature. It follows, that for systems employing a considerable proportion of direct water cooling, the water requirements vary appreciably as the outdoor temperature varies.

Generally, air conditioning systems are operated at full capacity when cooling is required, *i.e.*, as far as direct water cooling is concerned. When refrigeration is employed for further cooling, this part of the system is allowed to cycle thermostatically to provide temperature control, while the water part of the system is operated at maximum flow continuously without regard to cooling requirements. The refrigeration part of the system usually requires a much less quantity of water than the maximum design flow for the system. Inspection of the mean temperature curve indicates that for the greater part of the time that cooling is required, the outdoor temperature is less than maximum. At such times, a less than maximum flow should suffice.

If attention is given to means for regulating the flow of cooling water in accordance with load, a very substantial saving in seasonal water would result. This is true conservation, as the main concern is the quantity of water taken from the

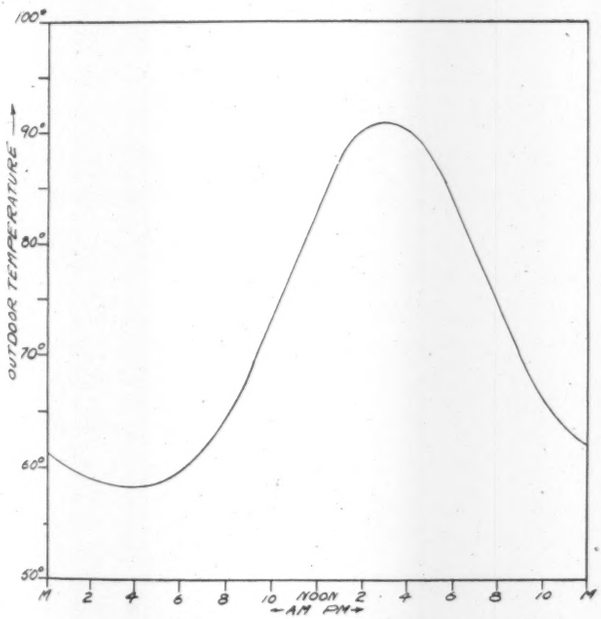

Fig. 1—Mean temperature curve for the months of June, July and August, 1940

(Plotted from data published by U. S. Department of Commerce Weather Bureau, Sacramento, Calif.)

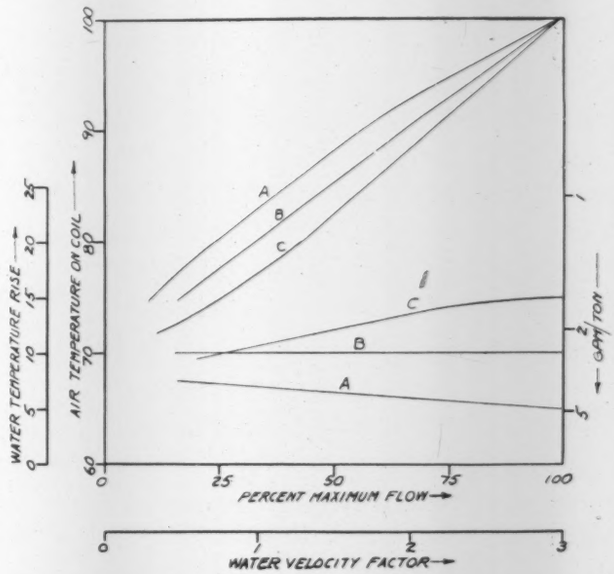

Fig. 3—Dry surface water coil characteristics under various operating conditions

ground over a period of time while the hourly or daily maximum demand is of little importance.

Water Cooling Under Varying Load Conditions

Fig. 2 is a plot of internal tube diameters as a function of water velocity for which Reynolds' Number is 2000; when Reynolds' number is 2000, or over, flow is turbulent and conventional heat transfer laws apply. It will be noted that for ½ in. to ¾ in. tubes, flow is turbulent for water velocities above ½ fps. Economic coil design for cooling with water at temperatures above the dew-point of the entering air requires a water velocity, at design maximum flow, in the order of 2.5 to 4 fps. It follows, that for normal application, the water velocity, *i.e.* flow, may be safely varied from design maximum down to 20 per cent maximum.

The curves of Fig. 3 show the per cent of design maximum water flow necessary to maintain a fixed delivery temperature at various entering air temperatures. Design conditions are 65 F entering water and a 5 F approach with entering air at 100 F. *Curves A, B,* and *C,* are for a design water temperature rise (hereafter indicated by DWTR) of 5 F, 10 F, and 15 F, respectively. The well-known formula is used:

H = U·MTD·A generally written,

H = Area × Rows depth × HTF × MTD

where HTF[1] (heat transfer factor) is expressed in Btu per square foot per degree MTD per row. The rate of change of HTF with respect to flow is nearly constant over a wide range of face velocities so the curves are typical for a wide choice of physical coil dimensions.

It is interesting to note that *Curve B* (Fig. 3), for a 10 F design water temperature rise, is a straight line over the operating range, *i.e.,* from 100 per cent flow to 20 per cent flow. Over the operating range, the flow drops off faster as the DWTR is reduced, and vice versa.

Curves plotted for a fixed DWTR at different values of design approach are similar in shape to those in Fig 3. In this case, the flow drops off faster as the design approach is increased. In general, the variation from the straight line relationship will be small so that it may be stated that the water requirements for fixed delivery temperature are approximately proportional to the coil loading.

At this point it is well to note that for most conditioning systems,[2] a fixed delivery temperature will at least maintain desired indoor to outdoor temperature difference. For most applications, and neglecting latent cooling,[3] if only 25 per cent of the internal load is proportional to the indoor and outdoor temperature difference, a fixed apparatus delivery temperature will maintain the following indoor to outdoor relation:[4]

Outdoor D. B.	Indoor D. B.
100	80
95	79
90	78
85	77
80	76
75	75

If more than 25 per cent of the internal load is proportional to indoor and outdoor temperature difference, the delivery temperature can be raised as the entering air temperature drops, and result with a faster falling off of the flow.

For systems using extensive amounts of water cooling, *i.e.,* designed for a close approach, and operating to furnish continuous air delivery with relatively high percentages of ventilation air (75-100 per cent), little cooling effect is gained by maintaining continuous water flow without regard to actual load, or, outdoor temperature. Substantial savings in total water requirements can be expected by modulating the water flow in accordance with load requirements.

The rate at which the coil water flow may be varied with respect to outdoor temperature is largely dependent on the percentage of ventilation air employed, the rate being highest at 100 per cent ventilation air.

Another consideration, which applies particularly to combination water and refrigeration systems, is minimum condenser requirements.[5] This would govern the minimum flow to which the water could normally be reduced, and would depend chiefly upon the percentages of design maximum load attributed to direct water cooling and mechanical refrigeration. For example, if the load ratio were 3 to 1 (water to refrigeration) and water cooling was effected at the rate of 2.4 gpm per ton (10 F DWTR), then if the design water temperature is 65 F and minimum condenser requirements 1.2 gpm per ton[6] with 75 F water, minimum percentage of maximum flow to which the water could be reduced would be:

$$\frac{1.2 \times 1}{2.4 \times 3} \cdot 100 = 16.66 \text{ per cent}$$

Fig. 4, *Curve A,* represents the total time for which given temperatures above 70 F are exceeded with respect to the total time above 70 F, expressed as per cent of time above 70 F. The curve is based on the average mean hours of 9:30 a.m. to 9:00 p.m. for the period noted (92 days × 11.5 hours = 1057 hours above 70 F). The curve is plotted from the mean temperature curve (Fig. 1) and rechecked from actual weather bureau records.[7] The curve is helpful in calculating the total water requirements for the period: 70 F is chosen as the base from the assumption that when the outdoor temperature is below this value ventilation alone will supply cooling or minimum flow only will be required. With this curve and a system water flow characteristic curve (such as Fig. 3) with a vertical line connecting the curve and the horizontal axis at the point of minimum flow, as shown in Fig. 5, the total water requirement for modulated flow may be estimated. For practical purposes, the curves may be divided into increments of Δ per cent flow and Δ per cent time respectively. A summation of the products of the two increments will be the water requirement for the period.[8]

An example is shown in Figs. 5 and 6. Fig. 5 is a per cent water flow curve for a typical system[9] as noted. Fig. 6 is a graphic check on total water requirement by plotting on the mean temperature curve

[1]HTF values used in calculating are as published by the Trane Co. for type *R* and *S* coils.

[2]Pertinent to this area.

[3]Loc. Cit. Note 2.

[4]Heating Ventilating Air Conditioning Guide, 1941, p. 62.

[5]It is assumed, in the light of conservation, that the water coil and condenser are series connected.

[6]Heating, Ventilating, Air Conditioning Guide, 1940, p. 498.

[7]For the months of June, July, and August, 1940.

[8]Similar to operating cost analysis, Heating, Ventilating, Air Conditioning Guide, 1941, pp. 471-472.

[9]Very typical in this area.

and using an arbitrary per cent water flow scale. For this example, the system operating under modulated water flow requires but 32 per cent of the water required over that if the system is operated at continuous flow whenever the outdoor temperature is above 70 F.

Curve B of Fig. 4 is per cent of time above 70 F for the average mean hours of 9:30 a.m. to 6:00 p.m. and applies to systems that are normally shut down at the close of the business day. The total operating hours for any increment of temperature above 70 F are found by multiplying the corresponding increment of per cent time by the total number of average mean hours for the period.

Comparison of Well with City Water

Pertinent data on city water for the major localities are listed in Table 2. For well systems, the main temperature is substantially constant throughout the year while for surface supply systems, the main temperature is subject to a rise as the summer season progresses.[10] City wells are carefully engineered and distributed so as to draw water over a relatively large area. From the standpoint of conservation, the use of city water is preferred to that from private wells where water rate, temperature, and disposal facilities are reasonably favorable.

The economic balance between well and city water is often in the direction of the latter when such factors as the cost of pumping, cost of well and pump, and proper water control with respect to cooling load, are duly appreciated.

The effective water rate, ER, corrected for cost of pumping may be written, where AR is the actual rate and rates are expressed in cost per 1000 gal[11]:

$$ER = AR - \frac{\text{head in feet} \times 0.00315 \times C}{\text{pump eff.} \times \text{motor eff.}}$$

where C is cost of power per KWH.

If modulated water flow with respect to load conditions is accomplished (as in examples of Figs. 5 and 6) the effective rate can be further reduced to values as low as 25 per cent of the actual rate, when compared to continuous maximum flow as would normally be the case for a well and pump installation.

Example: Fig. 5 is a water flow-outdoor temperature curve for a typical system with the following conditions: 100 per cent fresh air to be cooled at maximum design outdoor temperature of 100 F air to 60 F and to maintain a corresponding indoor temperature of 80 F. Water cooling effects 75 per cent of maximum load, i.e., from 100 F to 70 F using 2.4 gpm per ton. Air is then cooled at maximum load from 70 F to 60 F by refrigeration requiring 1.8 gpm per ton, or 25 per cent of maximum flow. Fifty per cent of the internal load is proportional to indoor-outdoor temperature difference and an assumption is made that the refrigeration part of the system will effect 10 F cooling at all operating conditions, the resultant air schedule:

Outdoor D. B.	Indoor D. B.	Off Water Coils	Entering Air	Per Cent Flow
100	80	70	60	100
95	79	71	61	73
90	78	72	62	45
85	77	73	63	25
80	76	74	64	25
75	75	75	65	25

Latent cooling load is neglected, or assumed to be absorbed by the refrigeration part of the system.

It can be seen that for temperatures below 85 F, 25 per cent flow is sufficient water. From the per cent of time curve (Fig. 4, *Curve* A) the outdoor temperature is at or below 85 F for 57.5 per cent of the total time above 70 F. Total time flow for this period is then 25 per cent flow × 57.5 per cent time, or a dimensionless value of 1437. (Minimum flow at times when the outdoor temperature is below 70 F is neglected). The other 42.5 per cent of the total time, the outdoor temperature is between 85 F and 100 F. For approximation, both curves are assumed to be straight lines and the average flow for this range is 57.5 per cent, and total time-flow 57.5 × 42.5 = 2444. Total time flow for period is then 2444 plus 1437 or 3881. This figure, compared to total time flow if the water were operated at maximum flow at total time above 70 F (100 per cent flow × 100 per cent time = 10,000) shows that the system requires but 38.8 per cent of the total water. Using smaller increments in temperature above 85 F would show a greater saving due to the abrupt rise in the per cent of time curve at higher outdoor temperatures.

Example: Fig. 6 is a graphic check on the described system; by plotting values of per cent flow at given outdoor temperatures on the mean temperature curve (Fig. 1), an arbitrary per cent flow scale is set up on the vertical axis. The large lined area (lined 30 F to left) represents the total time-flow if maximum flow were continuous at all time above 70 F as previously described; 100 per cent flow times average mean time of 11.5 hours (9:30 a.m. to 9:00 p.m.) or a dimensionless quantity of 1150. The smaller lined area (lined 30 F to right) is the time-flow for the system when flow is modulated. If the small hump above the 25 per cent flow line is assumed to be a sine curve, the area is easily integrated and becomes (2/pi) times the maximum height (25 per cent flow) times the base (5.5 mean hours) or 87. This added to 25 per cent flow for 11.5 hours gives a total time flow of 375. Compared to 1150, the total water requirement when the system is modulated is 33 per cent as compared to 38.8 per cent indicated in *Example* for Fig. 5.

Mechanical Methods for Control

The mechanical problem involved for modulating water flow in accordance with load requirements is one

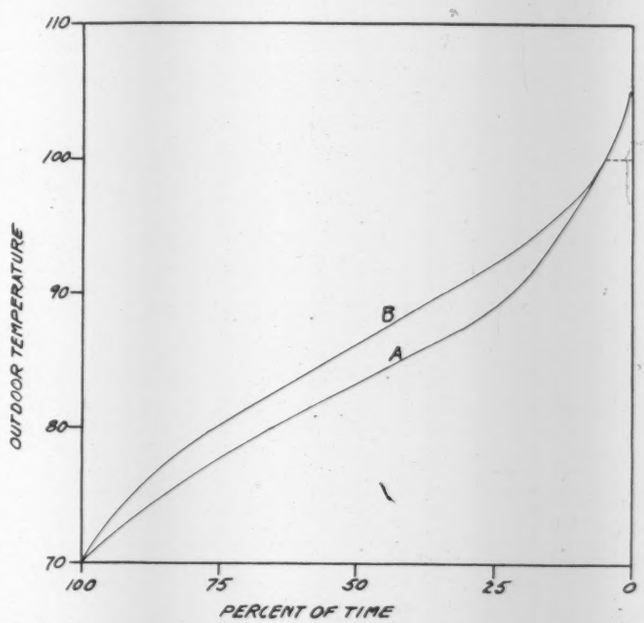

Fig. 4—Total per cent of time at temperatures above 70 F for months of June, July, and August, 1940, Sacramento, Calif.
Curve A. Based on average mean daily period from 9:30 a.m. to 9:00 p.m.
Curve B. Based on average mean daily period from 9:30 a.m. to 6:00 p.m.

[10]These points are substantiated in the Department of Commerce reports *Market Research Series*, Nos. 16, 17 for 1937 and 1938. Sacramento water main temperature does not agree with that published in the Heating, Ventilating, Air Conditioning Guide, 1941, Chapter 26.

[11]Cameron Hydraulic Data, 9th Edition.

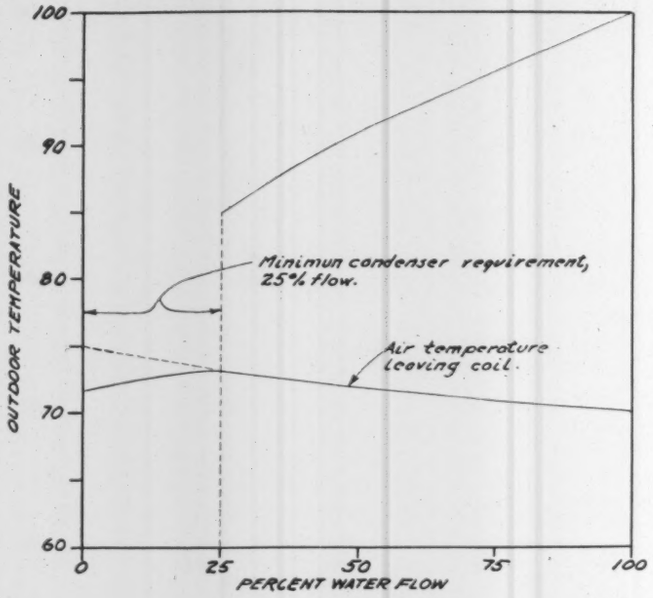

Fig. 5—Water flow, outdoor temperature curve for typical system

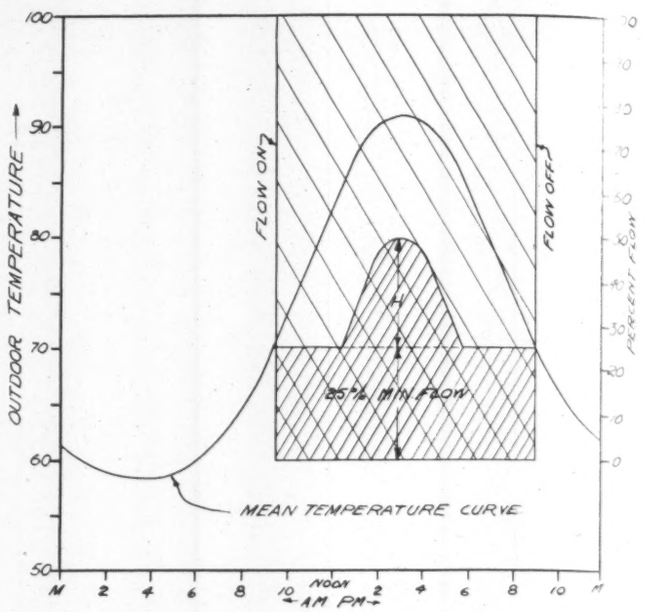

Fig. 6—Comparative water flow analysis for system of Fig. 5

of providing a suitable motor operated water valve having a *modulating* or proportioning type of motor operator inter-connected to an expedient thermostatic controller. The controller element can be located in the air stream leaving the coil and the circuit arranged so that the valve will throttle to positions that will admit just sufficient water necessary to maintain the desired leaving air temperature, or the controller element can be of the differential type and located in the air stream entering coil with the circuit arranged so that the valve will operate from the minimum open to fully open position as the entering air temperature varies from design minimum to maximum, and vice versa. Compensated control can be included to provide nearly any desired leaving-entering air temperature schedule.

Attention must be given to the valve port design, pressure drop at various positions, etc., so that the characteristics are suitable to the pressure variations in the water supply and cooling apparatus. If a pump is used for the source of supply, it must have characteristics suitable for operation over the required variations in discharge pressure, non-overloading. The problems are many and not discussed here; however, various types and makes of control equipment that will perform the required functions are commercially available.

Legislative Methods for Control

Limiting waste of cooling water to sewers is seemingly an effective method for controlling the underground supply within city limits, *i.e.* for an area supplied with a municipal sewer system. Like the Sacramento ordinance, division of limitation on a front of footage basis is deemed fair to all as to a certain extent average building size, relative taxation returns, etc., are taken into account.

There are, of course, many buildings with a small front footage to volume or cooling load ratio. In such cases, where the allowable disposal rate is insufficient for cooling needs, other heat removal devices such as return wells, cooling towers, or evaporative condensers are required. It is suggested that disposal might be limited on a square foot of property basis or developed property taxation basis; these bases would tend to effect a better balance and allow greater disposal for the larger buildings, not so dependent on a single factor such as front footage.

Another suggestion would be to limit disposal on a basis using either footage or taxation, with consideration for rate over a period of time, that is, limitation on an average hourly basis so that, with proper water control, greater quantities could be used at the hottest hours while the average over a period of time would be within reasonable limits. This would be true conservation from a supply water standpoint[12] and minimize the necessity for return wells. However, sewer capacity would become a deciding factor as all maximum flows from cooling systems would occur at the same time.

In some localities, legislation governing the installation of return wells may be required to prevent mixing between the supply and return strata, rise in well water temperature. Specifications as to depth,

Table 2—Source of City Water Supply

(Temperature at Supply—Average Maximum Water Temperature in Mains and Approximate Commercial Rate for Principal Municipalities in the Area)

City	Source	Temperature	Main Temperature	Rate per 100[a] cu ft
Modesto	Wells	68	69	$0.076
Stockton	Wells	69	70	$0.075
Lodi	Wells	66	67	$0.050
Sacramento	Surface[b]	—	72–77	$0.080
Marysville	Wells	66.5	67[c]	$0.075
Gridley	Wells	66.5	67	$0.075
Chico	Wells	64	64	$0.071

[a]Rate is expressed in cost per 100 cu ft and the average for a monthly consumption of 180,000 gal. The cost of the first 2000 cu ft is not included and considered normal domestic consumption.
[b]Sacramento is supplied by water from the Sacramento River which is subject to a wide variation in temperature throughout the year.
[c]Applies also to Yuba City.

[12]See second page of this paper, last paragraph.

casing material, etc., can be included in ordinances granting permission to dig wells under city sidewalks.

The possible increase in first cost, and/or operating cost that is effected for systems where restricting ordinances are enforced is in the true sense a small item compared to the probable results of allowing every commercial establishment to draw unlimited quantities of well water continuously.

Conclusions

The growth of air conditioning in the upper central and northern part of California presents problems to municipalities concerning the use and waste of cooling water. Means for regulation are desired for the protection of the underground supply, particularly in commercial areas.

Climatic conditions are such that a large percentage of the heat to be removed from buildings is in the form of sensible heat. The available cool water is used directly for cooling to a decided economic advantage and makes possible the use of high percentages of ventilation air.

The water rate for the cooling apparatus can be regulated in proportion to cooling load with the result of a substantial saving in total water required for the summer season. True conservation is then accomplished and in many cases the use of city water is warranted.

An effective method for controlling the underground supply in city areas is by providing ordinances limiting the rate of disposal to sewers. Limitation on a foot of frontage basis is used but other methods are suggested and require further study to provide ordinances which more equally distribute the use of water in accordance with cooling needs. Ordinances governing the use of disposal wells may be required in the near future.

The problems are different for each community and necessitate a comprehensive survey of the underground supply, city water facilities, disposal facilities, etc., to secure an ordinance which is best suited and fair to all.

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ST. LOUIS HOST TO DR. GIESECKE

March 4, 1941. The regular monthly meeting of the St. Louis Chapter was held at the David Ranken School and 80 members and guests attended. Following the reading of the minutes of the previous meeting, R. M. Rosebrough reported for the membership committee.

J. H. Carter, chairman of the program committee, announced that J. R. Vernon, Chicago, would be the speaker at the April meeting.

Pres. C. E. Hartwein, who presided at the meeting, then brought up the matter of a nominating committee for the Chapter election of officers to take place at the annual meeting of the Chapter, and the following were duly appointed members: Messrs. W. J. Oonk, G. W. F. Myers, E. E. Carlson, G. B. Rodenheiser, and R. J. Tenkonohy, the latter being selected as chairman. Messrs. L. R. Szombathy and M. F. Carlock were appointed by President Hartwein as Tellers of Election.

The secretary then read a press release sent to the Chapter from the Federal Government concerning a number of openings for technologists.

After a brief recess, the technical session convened in the auditorium of the School, at which time William Goodman, consulting engineer of The Trane Co., La Crosse, Wis., presented a most interesting and informative discussion on the subject of psychrometry. After discussion of the subject, the meeting adjourned.

February 6, 1941. The February meeting of the St. Louis Chapter was held at the Kingsway Hotel, with an attendance of 50 members and guests, including 14 ladies.

After a report on membership the treasurer submitted his report, followed by an announcement from the chairman of the program committee, who gave details of the March meeting.

Dr. and Mrs. F. E. Giesecke were the honored guests of the evening and were then presented by President Hartwein. Dr. Giesecke spoke briefly, thanking the Chapter for its hospitality and for the lovely flowers sent to Mrs. Giesecke.

At this time the supper meeting adjourned, and reconvened at the Engineers' Club, where a joint meeting was conducted with members of other engineering societies. Dr. Giesecke gave an interesting talk on Radiant Heating and Cooling, which was well received by his audience and caused considerable discussion.

ASHVE MEMBERS HONORED

W. H. Carrier, Chairman of the Board of Carrier Corp., Syracuse, N. Y., and R. M. Conner, Cleveland, O., director of the Testing Laboratories of the *American Gas Association*, were the recipients of medals awarded by the Franklin Institute, Philadelphia, Pa., at ceremonies at the Institute on May 21.

Dr. Carrier received the first Frank P. Brown Medal for improvements in the building and allied industries, and was selected for this honor because of his pioneer work in the creation and development of air conditioning.

Mr. Conner's award was for his services to the gas illumination, heat and power industry.

F. Paul Anderson Medal

The custom of recognizing special achievement by suitable award prevails in most technical and scientific societies. The F. Paul Anderson Medal is awarded in "recognition of notable scientific achievement or outstanding services performed in the field of heating, ventilating or air conditioning." Members of the American Society of Heating and Ventilating Engineers are requested to send nominations to Council for this important award.

E. O. Eastwood, *Chairman.*

Thermal Conductivity of Wood

By J. D. MacLean,* Madison, Wis.

INFORMATION on the thermal conductivity of wood and the influence of the more important variables affecting this property is of special interest from the standpoint of building insulation, the use of wood in connection with refrigeration, and in many other fields where the resistance of wood to heat transfer is a major consideration. This information is also of interest for comparing the thermal insulating properties of the different species of woods and for comparing wood with other insulating materials.

Previous studies on the heat conductivity of wood have been comparatively limited in extent from the standpoint of the number of species tested, variations in density within the same species, and the number of tests made on any given wood. Furthermore, the earlier experiments have been chiefly confined to wood having a relatively small moisture range. Often such investigations have been incidental to experiments on insulating products and fabricated structures such as wall sections.

The heat conductivity of wood is dependent on a number of factors of varying degrees of importance. Some of the more significant variables affecting the rate of heat flow in wood are the following: (1) density of the wood; (2) moisture content of the wood; (3) direction of heat flow with respect to the grain; (4) kind, quantity, and distribution of extractives or chemical substances in the wood, such as gums, tannins, or oils; (5) relative density of springwood and summerwood; (6) proportion of springwood and summerwood in the timber; (7) defects, like checks, knots, and cross grain structure.

The purpose of this paper is to report and discuss the results of a large number of heat conductivity experiments that have been made during the last 5 years at the Forest Products Laboratory† to determine the influence of some of the more important variables on thermal conductivity.

Preparation and Selection of Specimens: Specimens used in these experiments were selected from 32 species of wood, comprising both softwoods and hardwoods. A special effort was made to select specimens having different densities and different amounts of moisture within each species. Experiments were made on a number of species with the moisture content ranging from 0 (oven-dry) to that of wood in the green condition.

Two specimens were, of course, used in each run, each pair being matched as closely as possible by cutting them from the same board. The desired initial moisture content was obtained by storing the specimens at suitable constant temperature and humidity until they reached constant weight.

After the specimens reached constant weight in the conditioning room or oven, they were run through a planer and then cut to the required size and placed in test as quickly as possible, in order to avoid any appreciable change in moisture content. Care was taken to select specimens that were practically free from knots, checks, and other defects except in several runs that were made to investigate the influence of such defects.

The specimens were cut to dimensions of approximately 13½ × 13½ in. and the thickness generally ranged from about ½ to ¾ in. Thickness measurements were made with a micrometer to 0.001 in. Surface measurements (length and breadth) were made to the nearest 1/32 in.

A few specimens from species such as the southern pines, aspen, bald cypress, Engelmann spruce, and some of the oaks and maples, had a considerable amount of both sapwood and heartwood. The specimens of Douglas-fir, yellow birch, western red cedar, western larch, western hemlock, redwood, tanguile, prime vera, and white pine were all heartwood and the specimens from the other species were predominantly heartwood.

Many of the Douglas-fir specimens were cut from large diameter logs which made it possible to obtain specimens with small ring curvature. Several runs were therefore made on specimens of this species to compare the conductivity in the radial and tangential directions.

Test specimens of most of the other woods were cut from timbers of moderate diameter so that the direction of heat flow was partly radial and partly tangential. This combination of heat flow in the radial and tangential directions is, of course, more common in structural lumber than heat flow in either the radial or tangential direction alone. A few tests were also made to study the rate of heat flow in the longitudinal direction. These runs were made on Coast Douglas-fir and red oak.

Conditions of Test: In most of the experiments the specimens were kept under test for about 24 hours to allow the redistribution of moisture to take place and to make sure that constant temperature conditions were established. In a number of runs, the tests were continued for several days to observe the effect on moisture distribution and to find out whether the conductivity was appreciably changed during the longer period of test. In the latter runs the conductivity was determined at different time intervals varying from 24 to 48 hours during the period of test. In general, instrument readings for computation of conductivity were taken at frequent intervals for a period of about 1 hour or more and the results were averaged. No data were taken until the readings showed that constant conditions had been obtained.

Check runs were made at various times by letting the specimens cool to the same temperature on both sides without removing the test panels from the apparatus. The hot plate was then heated again and the conductivity test was repeated. In

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†Maintained at Madison, Wis., in cooperation with the University of Wisconsin.

these check runs the conductivity determined in the two cases was usually within about 1 per cent variation.

Most of the experiments were made with temperature differences of 40 to 60 F between the hot and cold plate surfaces. In some runs the conductivity was first determined with a temperature difference ranging between 40 and 60 F, and following this the hot plate temperature was raised to give a temperature difference of 80 to 90 F to investigate whether higher temperature differences (temperature of hot plate minus temperature of cold plate side) would make any important change in the conductivity. In most of the runs the temperature difference between the two faces of the specimens was about 53 F and the average temperature of the specimens (average temperature of hot plate and cold plate surfaces) was about 85 F.

In this paper the moisture content M is based on oven-dry weight

$$= \left(\frac{W-D}{D}\right) 100,$$

where W is the original weight and D is the weight after oven drying. The specific gravity S was always based on the oven-dry weight and the volume at current moisture content. In this case the oven-dry weight was taken as the original weight of the specimen divided by

$$\left(1+\frac{M}{100}\right)$$

where M is the percentage of moisture. On this basis the specific gravity

$$S = \left[\frac{W}{\left(1+\frac{M}{100}\right)}\right] \div (W_w)$$

$$= \frac{100W}{(100+M)\ (W_w)}$$

where W is the original weight of a unit volume of the wood at moisture content M, and W_w is the weight of an equal volume of water. For example, if W is in pounds per cubic foot, W_w is taken as 62.4 lb, the weight of 1 cu ft of water at maximum density.

Studies of the effect on moisture distribution caused by the difference in temperature between the hot and cold plate sides of the specimens were made. These tests comprised about 75 runs of different species having different amounts of water in the wood. The effect on moisture distribution when the wood was kept under test for various periods of time was also studied. In these experiments the following procedure was used in finding the moisture at different distances from the surface of the specimens. Immediately after test, a strip about ½ to ¾ in. wide and 2 or 3 in. long was cut from the central portion of each specimen and thin slices about ⅛ in. or less in thickness were cut from this strip with a mounted slicing knife. The slices were cut parallel with the faces of the test sections. Since the slices were very thin and not smooth enough for accurate measurement of thickness, the average effective thickness was determined from the proportion of the oven-dry weight of the specimen to the total oven-dry weight of all the sliced sections. The average moisture content was determined from moisture sections cut near the pieces which were prepared for slicing.

In engineering practice the conductivity K is commonly taken as the Btu that flow in 1 hour through 1 in. thickness of material 1 sq ft in area, when the temperature difference between the two surfaces is 1 F. This is the value of K shown for these tests and was computed from the formula:

$$K = \frac{Q\ x}{(T_1 - T_2)\ A\ \theta}$$

In this expression Q represents the quantity of heat expressed in Btu.

x = the thickness of the test panels in inches.
T_1 = the temperature on the hot plate side in degrees Fahrenheit.
T_2 = the temperature on the cold plate side in degrees Fahrenheit.
A = the area (expressed in square feet) through which the measured quantity of heat passes.
θ = the time in hours.

Table 1 presents a summary of the data obtained from 84 conductivity tests made on oven-dry specimens of 18 different species. The results of the tests on oven-dry wood showed that the conductivity was, in general, directly proportional to the specific gravity, over the range for the woods tested, as shown in Fig. 1. The plotted data in this curve were averaged for each 0.1 difference in specific gravity and the figures opposite the respective points show the number of runs in each of these group averages. The equation of the line in Fig. 1, showing the relation of conductivity K and specific gravity S for oven-dry wood, is

$$K = 1.39S + 0.165 \quad \ldots\ldots\ldots (1)$$

This may also be written, $K = 1.503S + 0.00165P$, where P is the percentage of air space and equals $100\ (1 - 0.685S)$. The constant 0.685 is the specific volume of oven-dry wood substance as determined in helium gas.

It may be noted that the straight line drawn through the plotted points in Fig. 1 passes through the point where the specific gravity S is zero and the conductivity K is 0.165. This value is the approximate conductivity of air at the average temperature of the test specimens. Fig. 2 shows the average computed conductivity values, using Equation (1), for each 0.1 difference in conductivity, plotted against the corresponding averages of the data determined in the experiments. The computed values of conductivity are indicated as K_1 in the tables. If the computed and test values were identical, the points would, of course, fall on a 45 deg line passing through 0. The close agreement of the computed and test values is shown by the fact that most of the plotted points fall on or very close to the 45 deg line.

The oak and maple specimens used in these runs contained a considerable proportion of sapwood with the exception of three runs that were made on all heartwood red oak. In these three runs on heartwood the computed conductivity averaged about 0.12 below the test values. This indicates that possibly chemical substances, such as tannin, increased the heartwood conductivity. The addition of 0.12 to the computed value for each of these three runs brings the last point plotted on Fig. 2 practically on the 45 deg line as is shown by the circle.

Species tested in the oven-dry condition had specific gravity values ranging from 0.11 to 0.76. Because of the cellular structure of wood, the relation between specific gravity and conductivity would theoretically be represented by a line gradually curving upward, assuming that the conductivity of wood substance is reasonably constant and the cell structure fairly uniform. The straight line, however, is a very close approximation for the range of specific

Table 1—Summary of Data from Conductivity Tests on Oven-Dry Wood

Species	No. of Runs in Average	Average Specific Gravity	Range of Specific Gravity	Average Conductivity From Test = K	Average Computed Conductivity[a] = K_1	Per cent Variation $\frac{(K_1-K)(100)}{K}$	Difference Between K_1 and K (K_1-K)	Remarks
Aspen, bigtooth (Populus grandidentata)	5	0.41	0.40 to 0.42	0.71	0.73	+2.8	+0.02	Heartwood & sapwood
Baldcypress (Taxodium distichum)	5	0.39	0.36 to 0.44	0.75	0.71	−5.3	0.04	Mostly heartwood
Balsa (Ochroma sp.)	4	0.16	0.11 to 0.21	0.41	0.38	−7.3	−0.03	
Basswood, American (Tilia glabra)	7	0.38	0.35 to 0.41	0.69	0.70	+1.4	+0.01	Heartwood & sapwood
Douglas-fir (Pseudotsuga taxifolia)	8	0.46	0.37 to 0.49	0.76	0.80	+5.3	+0.04	Heartwood
Elm, rock (Ulmus thomasii)	1	0.76	——	1.16	1.22	+5.2	+0.06	Heartwood & sapwood
Fir, white (Abies sp.)	2	0.41	0.40 to 0.42	0.71	0.74	+4.2	+0.03	Heartwood
Hemlock, western (Tsuga heterophylla)	2	0.46	0.44 to 0.49	0.79	0.81	+2.5	+0.02	Heartwood
Larch, western (Larix occidentalis)	3	0.57	0.52 to 0.61	0.94	0.95	+1.1	+0.01	Heartwood
Maple, sugar (Acer saccharum)	5	0.68	0.66 to 0.70	1.13	1.11	−1.8	−0.02	Heartwood & sapwood
Oak, red (Quercus sp.)	5	0.67	0.60 to 0.70	1.19	1.09	−8.4	−0.10	Heartwood & sapwood
Pine, southern yellow (Pinus sp.)	7	0.56	0.52 to 0.64	0.94	0.94	0.0	0.00	Heartwood & sapwood
Pine, white (Pinus sp.)	10	0.40	0.34 to 0.45	0.72	0.72	0.0	0.00	Mostly heartwood
Prima vera (Tabebuia bonnell-smithii)	1	0.47	——	0.82	0.82	0.0	0.00	Heartwood
Redcedar, western (Thuja plicata)	3	0.34	0.34 to 0.35	0.64	0.64	0.0	0.00	Heartwood
Redwood (Sequoia sempervirens)	8	0.40	0.32 to 0.51	0.74	0.72	−2.7	−0.02	Heartwood
Spruce, Engelmann (Picea engelmannii)	4	0.34	0.31 to 0.40	0.62	0.64	+3.2	+0.02	Mostly heartwood
Tanguile (Shorea polysperma)	4	0.58	0.57 to 0.59	1.00	0.97	−3.0	−0.03	Heartwood

[a] K_1 is conductivity computed from the equation $K_1 = 1.39S + 0.165$.

gravity values between 0 and 0.75. This range of specific gravity, based on oven-dry weight and volume when oven-dry, includes all the native softwoods and practically all the commonly used native hardwoods.

Table 2 shows data obtained in 385 tests on specimens of 32 different woods having varying amounts of water up to the fiber saturation point. In addition, data for 20 runs made on green specimens of 6 different species are shown in Table 3. The formulae for determining the computed conductivities shown in these tables are discussed under heading Computation of Conductivity.

Changes in Moisture Distribution: Studies of the moisture distribution after test showed that, although the moisture distribution was approximately uniform before test, there were often marked increases in the moisture content of the wood near the cold plate during the test period. These variations in moisture distribution are no doubt due to differences in vapor pressure produced by the differences in temperature. Their range was apparently influenced mainly by the original amount of water in the wood and the magnitude of the differences in temperature between the specimen faces. The rate at which equilibrium was approached was probably affected to a certain extent by the thickness and species.

Table 4 gives data on the moisture content of thin slices of wood about 0.07 to 0.15 in. thick taken from the warm and the cold side, respectively, of a considerable number of test specimens of different woods having various amounts of moisture. These are only part of the total runs made to study moisture distribution. All moisture data

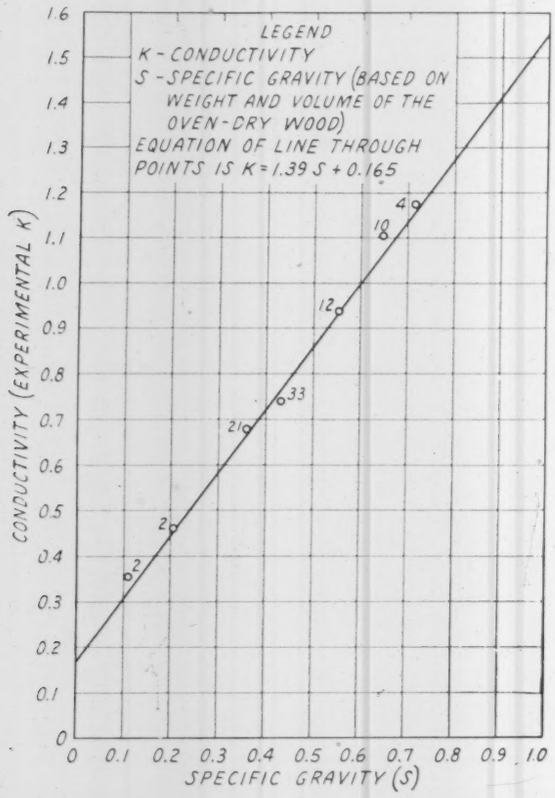
LEGEND
K - CONDUCTIVITY
S - SPECIFIC GRAVITY (BASED ON WEIGHT AND VOLUME OF THE OVEN-DRY WOOD)
EQUATION OF LINE THROUGH POINTS IS K = 1.39 S + 0.165
CONDUCTIVITY (EXPERIMENTAL K)
SPECIFIC GRAVITY (S)
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0 1.1 1.2 1.3 1.4 1.5 1.6

Fig. 1—Relation between conductivity determined in experiments on oven-dry wood, and specific gravity of specimens tested

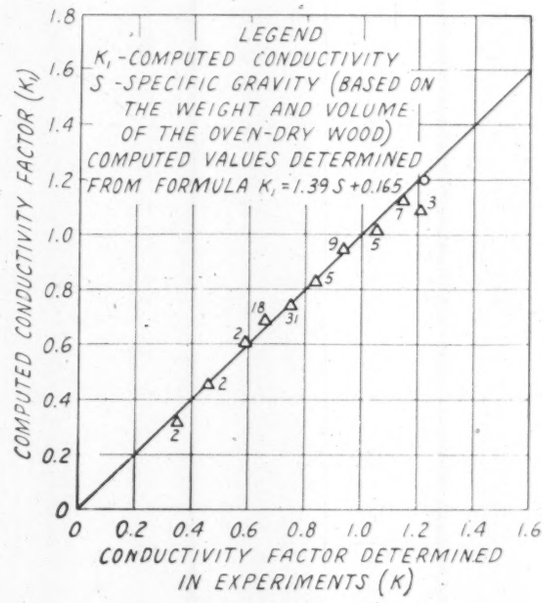
LEGEND
K1 - COMPUTED CONDUCTIVITY
S - SPECIFIC GRAVITY (BASED ON THE WEIGHT AND VOLUME OF THE OVEN-DRY WOOD)
COMPUTED VALUES DETERMINED FROM FORMULA K1 = 1.39 S + 0.165
COMPUTED CONDUCTIVITY FACTOR (K1)
CONDUCTIVITY FACTOR DETERMINED IN EXPERIMENTS (K)
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8

Fig. 2—Average computed conductivity plotted against average conductivity determined in tests on oven-dry wood

(Figures opposite points show number of runs in average. Data are averaged for each 0.10 difference in experimental conductivity factors. Circle shows position of last point when the computed values (K_1) for the three runs on heartwood of red oak had 0.12 added).

are the average values for the two specimens used in each run. Figs. 3 and 4 show representative moisture distribution curves for some of these runs. The data for each curve were obtained by slicing a cross section and determining the moisture content of each slice as previously explained. The results of tests not shown in Table 4 indicate that, when the average moisture content was about 10 per cent or less, the change in moisture distribution was comparatively small.

When the initial moisture content was about that of thoroughly air-seasoned wood (12 to 15 per cent) a large part of the redistribution of moisture usually occurred during the first 24 hours of test. This may be noted to be the general case from an examination of the data in Table 4 and is also shown by the moisture distribution curves for different periods of test. A few representative curves are given in Fig. 3. (See runs 90, 93, 99 and 100.) This early redistribution is somewhat obscured for some of the runs listed in Table 4 because the thickness of the slices from near the surfaces was more or less variable. The thinner slices would obviously show a higher moisture content near the cold surface and a lower moisture content near the hot-plate surface because the moisture content given is the average for the slice as taken. In addition, the wood in some specimens was more resistant to moisture movement than it was in others. Since this would affect the rate of moisture movement, the curves of moisture distribution for a given time interval will naturally show a certain amount of variability for different species and also to some extent within the same species.

When the initial moisture content was about 15 per cent or higher, it was noted after test that the wood surface in contact with the cold-plate was often damp or wet whereas that near the hot-plate side was at a moisture content ranging from about 7 to 12 per cent. Under such conditions some moisture naturally adhered to the metal surface of the cold plate. This amount of moisture was probably small but could not be determined.

Only a comparatively small change in moisture distribution occurred in a run on Douglas-fir specimens that had an average

Table 2—Summary of Data from Conductivity Tests on Specimens Having Various Amounts of Moisture Up to Fiber Saturation Point

Species	No. of Runs in Average	Average Specific Gravity	Range of Specific Gravity	Average Per cent Moisture	Range of Moisture Content Per cent	Average Conductivity from Test = K	Average Computed[a] Conductivity = K_1	Per cent Variation $\frac{K_1-K}{K}(100)$	Difference Between K_1 and K (K_1-K)	Remarks
Ash, white (Fraxinus americana)	7	0.56	0.55–0.57	15.6	12.1–17.2	1.21	1.19	— 1.7	—0.02	Mostly heartwood
Aspen, bigtooth (Populus grandidentata)	10	0.41	0.38–0.47	12.1	9.6–16.4	0.82	0.88	+ 7.3	+0.06	Heartwood & sapwood
Baldcypress (Taxodium distichum)	30	0.38	0.30–0.48	11.7	7.1–15.0	0.86	0.83	— 3.5	—0.03	Mostly heartwood
Balsa (Ochroma sp.)	6	0.17	0.10–0.21	8.0	3.2–10.4	0.47	0.44	— 6.4	—0.03	.....
Basswood, American (Tilia glabra)	20	0.37	0.33–0.41	9.3	3.8–11.5	0.74	0.77	+ 4.1	+0.03	Heartwood & sapwood
Birch, yellow (Betula lutea)	11	0.64	0.62–0.66	10.8	6.8–11.7	1.19	1.24	+ 4.2	+0.05	Heartwood
Douglas-fir (Pseudotsuga taxifolia)	52	0.46	0.41–0.51	18.4	7.2–33.2	0.97	1.03	+ 6.2	+0.06	Heartwood
Elm, rock (Ulmus thomasii)	9	0.65	0.59–0.69	12.9	12.1–31.2	1.34	1.38	+ 3.0	+0.04	Heartwood & sapwood
Greenheart (Ocotea rodioei)	4	0.85	0.84–0.87	18.4	16.5–19.9	1.90	1.79	— 5.8	—0.11	Heartwood
Hemlock, western (Tsuga heterophylla)	5	0.44	0.42–0.46	23.0	13.0–30.0	0.95	1.05	+10.5	+0.10	Heartwood
Larch, western (Larix occidentalis)	13	0.46	0.39–0.64	12.6	11.2–13.4	0.97	0.98	+ 1.0	+0.01	Heartwood
Maple, sugar (Acer saccharum)	27	0.66	0.62–0.72	11.7	5.2–27.8	1.40	1.31	— 6.4	—0.09	Heartwood & sapwood
Oak, red (Quercus sp.)	12	0.62	0.57–0.70	12.4	6.5–19.4	1.35	1.23	— 8.9	—0.12	Heartwood & sapwood
Oak, white (Quercus sp.)	18	0.62	0.55–0.74	11.1	5.4–20.1	1.37	1.23	—10.2	—0.14	Heartwood & sapwood
Pine, southern yellow (Pinus sp.)	59	0.53	0.43–0.71	13.8	5.2–23.8	1.12	1.11	— 0.9	—0.01	Heartwood & sapwood
Pine, white (Pinus sp.)	20	0.36	0.32–0.41	9.8	4.8–17.9	0.77	0.76	— 1.3	—0.01	Mostly heartwood
Prima vera (Tabebuia bonnell-smithii)	9	0.46	0.43–0.48	9.7	1.4–14.7	0.95	0.93	— 2.1	—0.02	Heartwood
Redcedar, western (Thuja plicata)	16	0.32	0.29–0.37	13.3	10.2–20.4	0.69	0.73	+ 5.8	+0.04	Heartwood
Redwood (Sequoia sempervirens)	15	0.39	0.30–0.48	11.7	6.8–12.7	0.82	0.83	+ 1.2	+0.01	Heartwood
Spruce, Engelmann (Picea engelmannii)	12	0.35	0.29–0.39	13.0	12.6–13.2	0.76	0.77	+ 1.3	+0.01	Heartwood & sapwood
Tanguile (Shorea polysperma)	6	0.54	0.53–0.56	10.6	6.0–14.2	1.05	1.08	+ 2.9	+0.03	Heartwood
Species From Which Material for Only One or Two Runs Was Available										
Beech, American (Fagus grandifolia)	1	0.59	——	11.1	——	1.17	1.17	0.0	0.00	Heartwood
Blackgum (Nyssa sylvatica)	1	0.46	——	10.0	——	1.00	0.93	— 7.0	—0.07	Heartwood & sapwood
Cherry, black (Prunus serotina)	1	0.68	——	7.8	——	1.21	1.26	+ 4.1	+0.05	Heartwood
Chestnut, American (Castanea dentata)	1	0.42	——	11.0	——	0.88	0.88	0.0	0.00	Heartwood
Elm, American (Ulmus americana)	1	0.54	——	9.4	——	0.98	1.06	+ 8.2	+0.08	Heartwood
Fir, white (Abies sp.)	2	0.38	0.37–0.38	11.7	10.8–12.0	0.78	0.81	+ 3.8	+0.03	Heartwood
Juniper, bigberry (Juniperus utahensis)	1	0.44	——	16.0	——	1.13	0.97	—14.2	—0.16	Heartwood & sapwood
Maple, silver (Acer saccharinum)	1	0.47	——	9.9	——	1.07	0.95	—11.2	—0.12	Heartwood
Pecan (Hicoria pecan)	1	0.67	——	10.0	——	1.38	1.28	— 7.2	—0.10	Heartwood
Sweetgum (Liquidambar styraciflua)	1	0.55	——	11.0	——	0.96	1.10	+14.6	+0.14	Heartwood & sapwood
Sycamore, American (Platanus occidentalis)	1	0.52	——	9.0	——	1.07	1.02	— 4.7	—0.05	Heartwood
Plywood Specimens										
Douglas-fir (Pseudotsuga taxifolia)	12	0.52	0.43–0.56	9.2	4.3–12.0	0.98	1.02	+ 4.1	+0.04	Heartwood

Correction for Infiltrated Materials

When 0.12 is added to computed conductivity K_1 for greenheart, for 14 runs on heartwood specimens of maple, and for all runs on heartwood specimens of red and white oak the following computed values are obtained which differ but little from the conductivity determined in the tests.

Greenheart (Ocotea rodioei)	4	.....	.....	.....	.....	1.90	1.91	+ 0.5	+0.01	.....
Maple, sugar (Acer saccharum)	27	.....	.....	.....	.....	1.40	1.37	— 2.1	—0.03	.....
Oak, red (Quercus sp.)	12	.....	.....	.....	.....	1.35	1.35	0.0	0.0	.....
Oak, white (Quercus sp.)	18	.....	.....	.....	.....	1.37	1.35	— 1.5	—0.02	.....

[a] K_1 is conductivity computed from the equation $K_1 = S\,[1.39 + 0.028M] + 0.165$.

moisture content of 16.4 per cent, when the specimens were under test only 5 hours. The curve of moisture distribution for this test is shown for run 92 in Fig. 3. Moisture distribution curves for runs 78 and 157 (Fig. 3) show that, when the initial moisture content was in the range of about 20 to 30 per cent, there was from 20 to 40 per cent difference at the end of the test, between the moisture content of the wood near the cold-plate and that near the hot-plate side. This is also shown for other runs, by data given in Table 4. It should be noted that the moisture content at the surface on the cold-plate side would be somewhat higher than the average shown in Table 4 for the thin section which included that surface.

A large proportion of the runs on these conductivity experiments were made on specimens in which the initial moisture content ranged from a little above zero to about the fiber-saturation point, because data on

Table 3—Summary of Data Obtained in Conductivity Tests on Green or Wet Specimens

Species	No. of Runs in Average	Average Specific Gravity	Range of Specific Gravity	Average Moisture Content, Per cent	Range of Moisture Content, Per cent	Average Conductivity from Test = K	Average Computed Conductivity[a] Thermal = K_1	Per cent Variation $\frac{(K_1-K)}{K}$ (100)	Difference Between K_1 and K (K_1-K)
Ash, white (Fraxinus americana)	1	0.48	——	91.1	——	2.60	2.49	—4.2	—0.11
Fir, white (Abies sp.)	3	0.36	0.33 to 0.38	107.7	77.8 to 129.8	2.10	2.13	+1.4	+0.03
Maple, sugar (Acer saccharum)	3	0.62	0.61 to 0.63	50.0	41.0 to 58.6	2.16	2.19	+1.4	+0.03
Oak, red (Quercus sp.)	10	0.56	0.54 to 0.59	60.3	36.0 to 82.0	2.23	2.24	+0.9	+0.02
Oak, white (Quercus sp.)	1	0.68	——	57.6	——	2.59	2.60	+0.4	+0.01
Redwood (Sequoia sempervirens)	2	0.36	——	80.2	71.7 to 88.9	1.89	1.76	—6.9	—0.13

[a] K is computed conductivity from equation $K = S[1.39 + 0.038\ M] + 0.165$.

Table 4—Comparison of Moisture Content in Wood Near Hot and Cold Plate Surfaces, Determined in Runs on Various Species

Species	Average Moisture Content of Specimens, Per cent	Moisture Content in Wood Near Hot Plate[a], Per cent	Moisture Content in Wood Near Cold Plate[a], Per cent	Time in Test[b], Days	Thickness of Specimens Used in Test, Inch	Average Specific Gravity	Temperature Difference Between Hot and Cold Plate Surfaces, F	Run Number
Ash, white (Fraxinus americana)	91.1	53.4	98.3	1	0.820	0.48	52	84
Aspen, bigtooth (Populus grandidentata)	15.0	9.4	24.6	1	0.842	0.40	55	97
Baldcypress (Taxodium distichum)	11.4	8.6	13.8	2	0.615	0.34	52	297
do.	12.2	9.8	16.2	1	0.711	0.39	59	120
do.	13.1	12.7	17.7	1	0.766	0.36	54	122
do.	13.6	9.4	17.6	5	0.760	0.32	53	105
do.	13.6	12.4	16.0	1	0.770	0.31	63	119
do.	14.0	13.4	16.2	1	0.736	0.31	56	111
Balsa (Ochroma sp.)	10.4	4.8	11.6	1	0.777	0.15	64	108
Basswood, American (Tilia glabra)	10.6	8.6	14.1	1	0.638	0.40	53	87
do.	11.5	7.6	11.8	1	0.606	0.39	51	86
Birch, yellow (Betula lutea)	11.2	9.2	11.2	1	0.785	0.63	48	60
do.	11.4	8.8	12.1	1	0.874	0.64	49	59
Douglas-fir (Pseudotsuga taxifolia)	10.2	8.6	10.8	1	0.660	0.47	52	91
do.	12.4	7.2	25.4	6	0.562	0.50	51	288
do.	13.0	9.5	15.2	4	1.495	0.47	53	294
do.	14.0	10.7	17.6	2	0.744	0.51	55	102
do.	14.0	9.3	23.8	5	0.684	0.50	48	101
do.	14.0	9.3	17.4	2	0.631	0.44	49	79
do.	14.0	8.6	15.6	1	0.628	0.45	49	94
do.	14.3	10.8	18.0	1	0.500	0.43	43	90
do.	14.4	8.3	18.1	3	0.608	0.45	50	93
do.	16.4	14.2	16.9	0.208 (5 hours)	0.628	0.46	48	92
do.	20.4	10.8	33.0	1	0.707	0.44	46	71
do.	21.9	12.4	31.8	1	0.709	0.44	46	106
do.	22.5	10.6	48.6	1	0.702	0.44	49	89
do.	27.7	12.6	47.0	1	0.713	0.48	47	77
do.	28.8	13.4	48.0	1	0.829	0.46	49	78
do.	34.6	8.0	102.2	3	0.716	0.49	63 and 69	303
do.	34.8	9.4	99.0	3	0.743	0.48	55 and 64	306
Fir, white (Abies sp.)	77.8	37.7	97.4	3	0.800	0.37	77	276
do.	115.4	106.8	122.5	3	0.737	0.38	42	258
Maple, sugar (Acer saccharum)	13.0	11.6	13.2	2	0.812	0.68	35	67
do.	13.6	11.4	13.4	1	0.750	0.68	38	69
do.	13.4	11.4	13.4	1	0.775	0.68	40	73
do.	13.7	9.0	19.2	5	0.795	0.69	49	98
do.	13.8	11.0	14.7	1	0.749	0.66	48	99
do.	13.9	12.0	14.0	1	0.810	0.69	40	72
do.	14.1	10.0	19.7	6	0.843	0.66	41	100
Oak, red (Quercus sp.)	58.4	16.6	68.1	4	0.795	0.55	66	285
do.	61.4	34.0	69.8	1¾	0.727	0.55	45	255
do.	64.2	33.7	82.0	3¾	0.750	0.56	41	254
do.	82.0	70.4	92.9	3	0.707	0.58	48	326
Pine, southern yellow (Pinus sp.)	11.8	7.6	14.8	1	0.707	0.53	50	152
do.	12.2	10.0	15.1	1	0.629	0.49	38	54
do.	13.1	9.0	15.7	1	0.652	0.49	43	53
do.	14.9	6.6	37.8	6	0.720	0.44	54	161
do.	17.6	11.4	33.0	1	0.808	0.54	47	160
do.	18.0	10.5	21.4	1	0.565	0.45	37	113
do.	18.2	12.6	31.8	1	0.804	0.50	50	157
Prima vera (Tabebuia bonnell-smithii)	7.4	6.7	7.6	1	0.808	0.43	51	104
do.	12.6	9.9	15.8	1	0.828	0.44	52	65
do.	13.3	9.5	13.4	1	0.808	0.46	42	66
do.	14.7	8.1	21.8	2	0.822	0.46	63	63
Redcedar, western (Thuja plicata)	12.1	7.6	18.6	1	0.725	0.30	53	57
do.	12.5	7.4	15.1	1	0.794	0.30	47	55
do.	16.4	10.0	27.6	1	0.652	0.31	59	121
do.	19.0	9.1	50.7	1	0.685	0.30	61	118
Spruce, Engelmann (Picea engelmannii)	12.8	6.6	34.2	3	0.692	0.31	66	171

[a] The moisture content values shown for the hot and cold plate sides are the averages found in thin slices varying from about 0.07 to 0.15 inch in thickness. The average moisture content of the specimens was determined from separate moisture sections cut near the sliced specimens.
[b] Each day in test was approximately 24 hours.

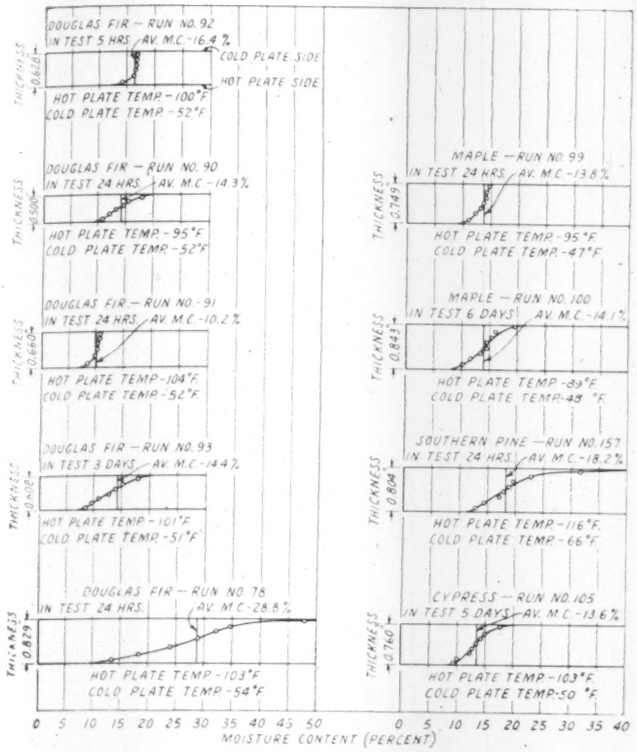
DOUGLAS FIR — RUN NO. 92
IN TEST 5 HRS. AV. M.C.-16.4 %
COLD PLATE SIDE
HOT PLATE SIDE
HOT PLATE TEMP.-100°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO. 90
IN TEST 24 HRS. AV. M.C.-14.3%
HOT PLATE TEMP.-95°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO. 91
IN TEST 24 HRS. AV. M.C.-10.2%
HOT PLATE TEMP.-104°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO. 93
IN TEST 3 DAYS AV. M.C.-14.4%
HOT PLATE TEMP.-101°F.
COLD PLATE TEMP.-51°F.
DOUGLAS FIR — RUN NO. 78
IN TEST 24 HRS. AV. M.C.-28.8%
HOT PLATE TEMP.-103°F.
COLD PLATE TEMP.-54°F.
MAPLE — RUN NO. 99
IN TEST 24 HRS. AV. M.C.-13.8%
HOT PLATE TEMP.-95°F.
COLD PLATE TEMP.-47°F.
MAPLE — RUN NO. 100
IN TEST 6 DAYS AV. M.C.-14.1%
HOT PLATE TEMP.-89°F.
COLD PLATE TEMP.-49°F.
SOUTHERN PINE — RUN NO. 157
IN TEST 24 HRS. AV. M.C.-18.2%
HOT PLATE TEMP.-116°F.
COLD PLATE TEMP.-66°F.
CYPRESS — RUN NO. 105
IN TEST 5 DAYS AV. M.C.-13.6%
HOT PLATE TEMP.-103°F.
COLD PLATE TEMP.-50°F.
THICKNESS
0 5 10 15 20 25 30 35 40 45 50
0 5 10 15 20 25 30 35 40
MOISTURE CONTENT (PERCENT)

Fig. 3—Moisture distribution curves for seasoned or partially seasoned wood, after different periods in test

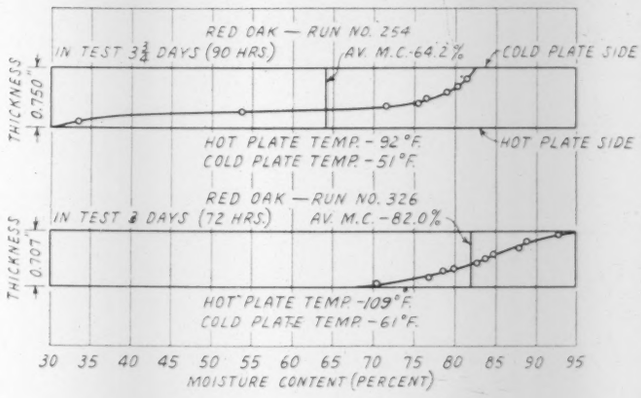
RED OAK — RUN NO. 254
IN TEST 3¾ DAYS (90 HRS.) AV. M.C.-64.2% COLD PLATE SIDE
HOT PLATE TEMP.-92°F.
COLD PLATE TEMP.-51°F.
HOT PLATE SIDE
RED OAK — RUN NO. 326
IN TEST 3 DAYS (72 HRS.) AV. M.C.-82.0%
HOT PLATE TEMP.-109°F.
COLD PLATE TEMP.-61°F.
THICKNESS
0.750
0.707
30 35 40 45 50 55 60 65 70 75 80 85 90 95
MOISTURE CONTENT (PERCENT)

Fig. 4—Moisture distribution curves for green red oak after different periods in test

conductivity are of principal interest in this range. Table 5 shows the number of runs made on each species for moisture content intervals of 5 per cent up to a little above the fiber-saturation point. All runs made on specimens having an average moisture content over 40 per cent are placed in one group and are classed as green.

Effect of Different Periods of Test on Conductivity: Several runs were made to study the effect of different periods of test on the conductivity since it was recognized that the final moisture distribution was dependent to a certain extent on the time the wood was exposed to temperature differences between the two faces. Data on these tests are given in Table 6. The results indicate that when the average moisture content was about that of air-seasoned wood, 15 per cent or less, there was usually no significant change in conductivity after 24 hours in test. In fact, data taken 5 or 6 hours after specimens having this range of moisture content were placed in test, showed there was little change and often no change in conductivity from that found after test periods of 24 hours or more.

When the average moisture content was in the range of 20 to 35 per cent there was some tendency for the conductivity to decrease with increase in time of test. This is shown in data for runs 290, 303 and 306 in Table 6. This change would probably continue until a condition of moisture equilibrium was established for the differences in temperature between the two faces. It is very probable that a considerable part of the redistribution of moisture may have occurred before the first readings of conductivity were taken. For this reason the first determination of conductivity is not necessarily the maximum conductivity.

Green wood having a moisture content of about 60 per cent or more showed a much slower trend toward a lower conductivity than specimens having a lower moisture content close to the fiber-saturation point. This slow change found for the wet wood may be on account of the smaller amount of air space in the wood cells because of the large amount of water present. The moisture distribution curves for runs 254 and 326 (Fig. 4) show that even after 72 to 90 hours in test (3 to 3¾ days) the moisture content of the wood next to the hot plate was not below the fiber-saturation point.

Results of these tests show that, since the redistribution of moisture has a tendency to reduce rather than increase the conductivity, one can logically assume that the conductivity data will represent maximum values that may be expected for the particular average moisture content, regardless of the time required to establish equilibrium moisture conditions. While it is recognized that the conductivity as determined does not represent the conductivity of the wood under the original conditions of moisture distribution, even if it were possible to devise a method of finding the conductivity without disturbing the original moisture balance, such data would not represent conditions commonly encountered. A moisture gradient necessarily occurs when wood containing moisture is used in any type of structure where a temperature gradient exists between faces. Since specimens tested in the hot-plate apparatus do not evaporate moisture from the cold surface, the average moisture content will not change with change in moisture distribution except for a very slight moisture loss at the edges that does not affect the interior portion. Whether moisture evaporates from the cold side when wood is used commercially depends upon a variety of factors.

Since both the specific gravity and moisture content can be determined or estimated within reasonable limits for a given species and for particular conditions of service under which the wood is used, a formula for computing the approximate conductivity would be of particular assistance in questions relating to the insulating properties of different woods. An equation showing the relation of conductivity, specific gravity, and moisture content would make it possible to disregard species and to calculate, within reasonable limits, the con-

Table 5—Range of Moisture Content of Test Specimens of Different Species

Species	Number of Runs Made at Moisture Content of								Total No. of Runs on Each Species
	Zero or Oven Dry	Between Zero and 5 Per cent, Incl.	Between 5 + and 10 Per cent, Incl.	Between 10 + and 15 Per cent, Incl.	Between 15 + and 20 Per cent, Incl.	Between 20 + and 25 Per cent, Incl.	Between 25 + and 30 Per cent, Incl.	Green or Wet Wood (Over 40 Per cent)	
Ash, white (Fraxinus americana)	—	—	—	1	6	—	—	1	8
Aspen, bigtooth (Populus grandidentata)	5	—	1	8	1	—	—	—	15
Baldcypress (Taxodium distichum)	5	—	6	24	—	—	—	—	35
Balsa (Ochroma sp.)	4	1	4	1	—	—	—	—	10
Basswood, American (Tilia glabra)	7	2	9	9	—	—	—	—	27
Birch, yellow (Betula lutea)	—	—	1	10	—	—	—	—	11
Douglas-fir (Pseudotsuga taxifolia)	8	—	3	27	1	6	15	—	60
Elm, rock (Ulmus thomasii)	1	—	—	5	2	—	2	—	10
Fir, white (Abies sp.)	2	—	—	—	—	—	—	3	5
Greenheart (Ocotea rodioei)	—	—	—	—	4	—	—	—	4
Hemlock, western (Tsuga heterophylla)	2	—	—	1	—	2	2	—	7
Larch, western (Larix occidentalis)	3	—	—	13	—	—	—	—	16
Maple, sugar (Acer saccharum)	5	—	7	19	—	—	1	3	35
Oak, red (Quercus sp.)	5	—	1	10	1	—	—	10	27
Oak, white (Quercus sp.)	—	—	9	5	3	1	—	1	19
Pine, southern yellow (Pinus sp.)	7	—	2	43	7	7	—	—	66
Pine, white (Pinus sp.)	10	2	5	11	2	—	—	—	30
Prima vera (Tabeuia bonnell-smithii)	1	3	1	5	—	—	—	—	10
Redcedar, western (Thuja plicata)	3	—	—	12	3	1	—	—	19
Redwood (Sequoia sempervirens)	8	—	1	14	—	—	—	2	25
Spruce, Engelmann (Picea engelmannii)	4	—	—	12	—	—	—	—	16
Tanguile (Shorea polysperma)	4	—	2	4	—	—	—	—	10
Miscellaneous species[a]	—	—	6	5	1	—	—	—	12
Plywood Specimens									
Douglas-fir (Pseudotsuga taxifolia)	—	1	5	6	—	—	—	—	12
Total									489

[a]Species on which only one or two runs were made.

ductivity of any wood without making conductivity tests.

Published data frequently give the conductivity and weight per cubic foot for a particular species, but no data are given regarding the moisture content at time of test. Since differences in the specific gravity of wood have a very different effect on conductivity from that caused by differences in the amount of water in the wood, it is evident that the weight per unit volume might be the same but the conductivity could vary considerably, depending on the proportional weight of water and wood substance. Although it is not practicable to compute the exact conductivity of wood of given density and moisture content because of the number of variables involved, the results of the experiments discussed in this paper indicate that it is possible to compute the conductivity closely enough for practical purposes.

In deriving an expression for the relation of conductivity, air space, moisture content, and specific gravity, it has been assumed that the conductivity of water-free wood is represented by the formula

$$K = 1.503\ S + \left(\frac{P}{100}\right)(0.165)$$

$= 1.39S + 0.165$ as shown in Equation (1). In order to find an expression for computing the conductivity of wood containing water it was assumed that another term MSX could be added where $MS =$ the percentage of volume occupied by water and $X =$ a factor representing the effect of moisture on conductivity. The equation for the conductivity of wood containing water could then be written:

$$K = 1.503S + \frac{P}{100}(0.165) + MSX \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Substituting for $\frac{P}{100}$ its value $\left[1 - S\left(0.685 + \frac{M}{100}\right)\right]$

Table 6—Conductivity of Wood After Different Periods of Heating

Species	Average Moisture Content Per Cent	Time in Test Hours	Conductivity K	Run No.	Remarks
Baldcypress (Taxodium distichum)	13.6	24	0.72	105	
		96	0.73		
		120	0.74		
Baldcypress (Taxodium distichum)	11.4	24	0.82	297	
		48	0.82		
Douglas-fir (Pseudotsuga taxifolia)	14.0	24	0.92	101	
		96	0.90		
		120	0.91		
Douglas-fir (Pseudotsuga taxifolia)	14.0	24	0.97	102	
		48	0.95		
Douglas-fir (Pseudotsuga taxifolia)	12.4	2	0.90	288	
		20	0.85		
		44	0.87		
		113	0.85		
		144	0.83		
Douglas-fir (Pseudotsuga taxifolia)	34.6	24	1.11	303	Specimens soaked in water about 6 weeks before test
		72	0.99		
Douglas-fir (Pseudotsuga taxifolia)	34.8	24	1.21	306	
		72	1.02		
Maple, sugar (Acer saccharum)	14.1	24	1.58	100	
		96	1.59		
		120	1.56		
Maple, silver (Acer saccharinum)	9.9	24	1.07	464	
		48	1.06		
Oak, red (Quercus sp.)	64.2	60	2.32	254	
		90	2.25		
Oak, red (Quercus sp.)	66.7	24	2.23	257	
		48	2.24		
Oak, red (Quercus sp.)	58.4	24	2.20	285	
		96	2.10		
Oak, red (Quercus sp.)	82.0	18	2.88	326	Soaked in water about one month before test
		72	2.87		
Pine, southern yellow (Pinus sp.)	14.9	24	1.01	161	
		96	1.01		
		120	1.00		
		144	0.98		
Pine, southern yellow (Pinus sp.)	23.8	24	1.24	290	
		120	1.13		
Redwood (Sequoia sempervirens)	71.7	24	1.66	314	Soaked in water about 6 weeks before test
		48	1.66		

Equation (2) becomes, $K = S [1.39 + M (X - 0.00165)] + 0.165$.

If Z represents the term $(X - 0.00165)$

$$K = S [1.39 + MZ] + 0.165 \ldots\ldots (3)$$

The factor Z will necessarily vary within a certain range, partly because the moisture distribution is not uniform and also because of variability in the wood structure, extractives, and other factors that affect the conductivity of oven-dry wood. For moisture content values below the fiber-saturation point (taken as 30 per cent), the majority of the species tested had a value of Z ranging between 0.018 and 0.039 and the average for the different woods with a moisture content range up to the fiber-saturation point was about 0.028. The average value for Z determined for the tests on specimens having from 50 to over 100 per cent moisture was 0.038.

There is, of course, no strict demarcation for the value of Z at the fiber-saturation point. In general, Z should probably increase somewhat gradually up to a maximum for green material. The equations are necessarily based on average values for both oven-dry wood and for wood containing varying amounts of water. For moisture-content values over 30 but under 40 per cent, it would probably be a little closer to use the factor 0.028 in computing K even though 40 per cent moisture content is about 10 per cent above the fiber-saturation point, taken as 30 per cent.

Substituting 0.028 and 0.038 in Equation (3) for the respective groups gives:

$$K = S (1.39 + 0.028M) + 0.165 \ldots\ldots (4)$$

for wood having varying amounts of moisture up to a little over the fiber-saturation point (values under 40 per cent); and

$$K = S (1.39 + 0.038M) + 0.165 \ldots\ldots (5)$$

for green wood having 40 per cent or more, moisture.

In the foregoing expressions, K = the conductivity, M = the moisture content, and S = the specific gravity as previously defined. It is evident that if the value of $M = 0$ in Equations (4) and (5), both become the same as Equation (1) derived for oven-dry wood.

The difference in the computed and the experimental value of conductivity for partly seasoned wood was highest for heartwood specimens of red and white oak, greenheart, and maple. Three runs made on oven-dry heartwood specimens of red oak and one run on maple specimens, mostly heartwood, also showed somewhat higher conductivities, considering the specific gravity, than the other woods. On the other hand, runs made on heartwood specimens of partly seasoned rock elm, white ash and yellow birch, which are also relatively heavy woods, did not show the difference in computed and test conductivity values noted for heart material of the other woods named.

Some of the maple specimens had a considerable amount of sapwood and the values of the computed and experimental conductivity factors for these specimens were fairly close. It appears possible that extractives, such as tannin in the oaks, and various substances found in the heartwood of other species may be largely responsible for the higher conductivity of the woods in question. The difference between the computed value of K and that found from the tests on the heartwood specimens of oak, greenheart, and maple was fairly constant and was in the neighborhood of 0.12. If this difference is added to the computed conductivity for heartwood specimens, the test values and computed values of K for these four woods will be very close. Only the heartwood specimens of these woods, however, showed this amount of variation. In a few runs on heartwood specimens of oak the computed conductivity checked very closely with the conductivity found in the tests but more commonly the computed value was a little lower.

The effect of extractives in green oak and maple was apparently obscured by the high moisture content of the specimens since the computed values of K were, in general, very close to those determined in the tests and, as shown in Table 3, varied less than 5 per cent.

Cypress showed a more conspicuous variation in conductivity than many of the other woods tested and it is probable that extractives, such as the oil in the wood, affect the results in this case.

Douglas-fir was more variable than the southern pines and, in general, had a trend toward a little lower conductivity than other woods of similar specific gravity and moisture content. It was observed that the oven-dry specimens had somewhat lower conductivity for wood of that specific gravity, indicating that there was something inherent in the wood structure affecting the conductivity of this species. The tests on southern pine included a number of runs on resinous wood, but the increase in conductivity was, in most cases, in about the same proportion as the increase in specific gravity because of the presence of resin. Apparently the conductivity of the resin was not much different from that of the wood substance.

Fig. 5 shows the average computed conductivity plotted against the average conductivity determined in test for specimens having a moisture content ranging from about 2 to 33 per cent. The values are averaged for each 0.1 difference in conductivity. Fig. 6 shows similar data plotted for the tests on green specimens. If the average values computed and the average values determined in the experiments were the same, the plotted points in Figs. 5 and 6 would, of course, fall on the 45 deg line. It may be noted that most of the points in Fig. 5 with the exception of the last few near the upper end of the line, fall on or very close to the 45 deg line. The last points tend to diverge, mainly because of the relatively higher conductivity found in the heartwood of the greenheart, oak and maple specimens. When 0.12 is added to the computed conductivity for these woods the points fall much closer to the line, as shown by the small circles.

Table 3 shows that the computed and test values for the green material are very nearly the same. In these runs on green wood, the moisture content was 50 per cent or more and the factor 0.038 was used for Z in making the computations.

The data in Table 7 give the variations between the computed conductivity for the different species and the values found in the tests when the moisture content range was from 0 to approximately the

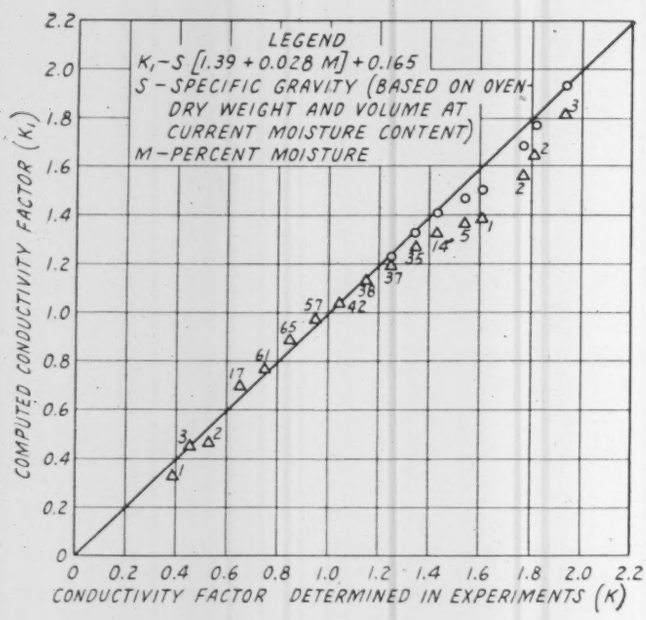
LEGEND
$K_1 = S\,[1.39 + 0.028\,M] + 0.165$
S – SPECIFIC GRAVITY (BASED ON OVEN-DRY WEIGHT AND VOLUME AT CURRENT MOISTURE CONTENT)
M – PERCENT MOISTURE
COMPUTED CONDUCTIVITY FACTOR (K_1)
CONDUCTIVITY FACTOR DETERMINED IN EXPERIMENTS (K)

Fig. 5—Average computed conductivity plotted against average conductivity determined in tests on wood with different amounts of moisture

(Circles show averages when 0.12 is added to computed values (K_1) for heartwood of greenheart, sugar maple and oak. Numbers opposite points show number of runs in average. Data are averaged for each 0.10 difference in experimental conductivity factors.)

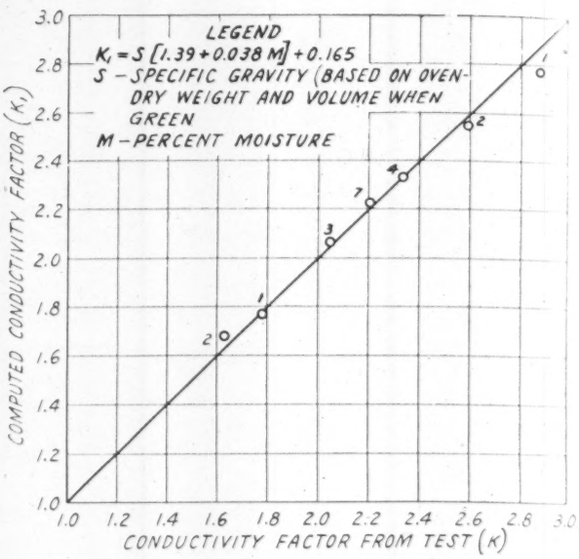
LEGEND
$K_1 = S\,[1.39 + 0.038\,M] + 0.165$
S – SPECIFIC GRAVITY (BASED ON OVEN-DRY WEIGHT AND VOLUME WHEN GREEN
M – PERCENT MOISTURE
COMPUTED CONDUCTIVITY FACTOR (K_1)
CONDUCTIVITY FACTOR FROM TEST (K)

Fig. 6—Average computed conductivity plotted against average conductivity determined in tests on green wood

(Data averaged for each 0.20 difference in experimental conductivity factors. Numbers opposite points show number of runs in average.)

fiber-saturation point. The data in this table show that in the total of 84 runs made on oven-dry specimens 57 per cent of the computed conductivity values were within 5 per cent of those determined in the tests; about 29 per cent more were between 5 and 10 per cent; and less than 4 per cent were in the range of maximum variation, which was between 15 and 20 per cent of the values found in the tests.

In the group of 385 runs made on specimens with varying amounts of moisture up to a maximum of about 33 per cent, about 42 per cent of the computed values were within 5 per cent of the test values; 33 per cent more were between 5 and 10 per cent; 20 per cent were between 10 and 15 per cent; and less than 2 per cent were in the range of maximum variation of 20 to 25 per cent. If 0.12 is added to the computed values of conductivity for the heartwood specimens of red and white oak, maple, and greenheart about 50 per cent of the computed conductivity values are within 5 per cent of the values found in the tests; about 31 per cent within 5 and 10 per cent; and 14 per cent between 10 and 15 per cent of the conductivity determined in the tests. Altogether, about 95 per cent of the computed conductivity values varied less than 15 per cent from the values found in the tests on wood having varying amounts of moisture up to the fiber-saturation point. Nearly 80 per cent varied less than 10 per cent from the experimental values.

Fig. 7 shows the relation of conductivity, moisture content, and specific gravity for moisture values ranging from 0 to 30 per cent. The specific gravity and moisture content can be easily determined as previously explained under the subheading, *Conditions of Test*. It is often more convenient to determine the original weight of wood per unit volume instead of the specific gravity; the original weight being the weight of the wood and the water it contains at the current moisture content, M. Fig. 8 was therefore prepared to show the relation of weight of wood and conductivity for moisture content intervals ranging from 0 to 30 per cent. The original weight of wood in pounds per cubic foot (actual dimensions) is shown on the bottom scale and the corresponding weight in pounds per square foot and 1 in. in thickness is shown at the top. The approximate average moisture content must of course be known or assumed in order to use this chart. The data for Figs. 7 and 8 were computed by means of Equation (4). Fig. 8 illustrates the point that data showing the original weight per unit volume and conductivity only, are not sufficient since the weight will depend both on specific gravity and moisture content. For this reason either the moisture content or the specific gravity S should also be given in addition to the weight per unit volume. Either Fig. 7 or 8 will be found convenient for finding the approximate conductivity of any particular species, or the conductivity factor can be computed by means of Equation (4) when the moisture content is under 40 per cent. Equation (5) can be used in computing the approximate conductivity of green wood when the moisture content is about 40 per cent or higher.

When wood seasons from the green condition to any given moisture content below the fiber-saturation point, the reduction in moisture will tend to reduce the heat conductivity. On the other hand, the specific gravity will increase because of shrinkage in seasoning and this will tend to increase the conductivity. Very often it is desirable to compare the conductivity of one species with that of another when the wood has a particular moisture content, and it is also of interest to know how the conductivity of a given wood changes in going from the green to the oven-dry condition.

Shrinkage studies made at the Forest Products Laboratory show that the percentage volumetric shrinkage (Vs) that occurs in seasoning from the fiber-saturation point to any moisture content below the fiber-saturation point is approx-

imately proportional to the loss of water from the cell walls. Although some woods having the same specific gravity based on volume when green and weight when oven-dry may not shrink to quite the same extent between the green and oven-dry conditions, for most species the differences in shrinkage are usually small when the specific gravity of the green wood is the same. It is therefore sufficiently accurate for practical purposes to compute the shrinkage and the corresponding specific gravity on the basis that shrinkage is proportional to the loss of water from the cell walls.

The approximate volumetric shrinkage would then be computed from the relation:

$$Vs = (Sg)\left[\left(\frac{1}{1.115}\right)(30 - M)\right]$$

or since $\frac{1}{1.115} = 0.9$, approximately

$$Vs = Sg\ [0.9\ (30 - M)]$$

In this expression Vs is the percentage volumetric shrinkage; Sg is the specific gravity based on volume when green and weight when oven-dry; 1.115 is the density of the water in the fiber walls when they are saturated and M is the percentage moisture to which the wood is seasoned.

If Sm represents the specific gravity after seasoning to a moisture content M the value of Sm is found from the relation

$$Sm = \left[\frac{Sg}{1 - \frac{Vs}{100}}\right] = \frac{Sg}{1 - Sg\ [0.009\ (30 - M)]}$$

Likewise Sg can be computed if Sm is known or assumed since

$$Sg = \frac{Sm}{1 + (Sm)\ [\ (0.009)\ (30 - M)\ ]}$$

Average values of Sg for a considerable number of the more important commercial native species are given in Table 8. Similar data for other woods are given in U.S.D.A., Tech. Bul. 479. Since seasoned wood soaked in water until it is thoroughly wet will swell to about green dimensions, the value of Sg can be readily determined for samples from any timber regardless of the degree of seasoning, by assuming the green volume to be the same as the volume of the sample after soaking to maximum swelling in water.

If the computed specific gravity Sm is substituted for S in the equation $K = S\ (1.39 + 0.028M) + 0.165$ the approximate conductivity K can be computed for wood at any given moisture content using the specific gravity Sg based on the green wood. Substituting for Sm its equivalent

$$\left[\frac{Sg}{1 - \left(\frac{1\ s}{100}\right)}\right]$$

the equation for conductivity becomes:

Table 7—Variation of Computed Values of Conductivity from Values Found in Tests (*Moisture Content Range 0 to Fiber Saturation Point*)

Species	No. of Runs in Which Computed Conductivity is Within the Following Percentage Variation From Value of K Determined in Experiments					No. of Runs on Oven-Dry Wood	No. of Runs on Wood Containing Moisture
	0 to 5 Per Cent	5 + to 10 Per Cent	10 + to 15 Per Cent	15 + to 20 Per Cent	20 + to 25 Per Cent		
Ash, white (Fraxinus americana)							
(Oven-dry wood)	—	—	—	—	—		
(Wood having 12-18 per cent moisture)	6	—	1	—	—		7
Aspen, bigtooth (Populus grandidentata)							
(Oven-dry wood)	2	2	1			5	
(Wood having 9-17 per cent moisture)	3	3	2	2			10
Baldcypress (Taxodium distichum)							
(Oven-dry wood)	1	3	—	1		5	
(Wood having 7-15 per cent moisture)	6	12	4	4	4		30
Balsa (Ochroma sp.)							
(Oven-dry wood)	1	2	1			4	
(Wood having 3-10 per cent moisture)	3	1	—	2			6
Basswood, American (Tilia glabra)							
(Oven-dry wood)	7					7	
(Wood having 3-12 per cent moisture)	7	9	4				20
Birch, yellow (Betula lutea)							
(Oven-dry wood)	—	—	—	—	—		
(Wood having 6-12 per cent moisture)	6	4	1				11
Douglas-fir (Pseudotsuga taxifolia)							
(Oven-dry wood)	1	4	2	1		8	
(Wood having 7-33 per cent moisture)	14	14	19	4	1		52
(Plywood having 4-12 per cent moisture)	5	1	4	2			12
Elm, rock (Ulmus thomasii)							
(Oven-dry wood)	1	—	—	—	—	1	
(Wood having 12-32 per cent moisture)	6	3	—	—	—		9
Fir, white (Abies sp.)							
(Oven-dry wood)	1	—	1			2	
Greenheart[a] (Ocotea rodioei)							
(Oven-dry wood)	—	—	—	—	—	—	
(Wood having 16-20 per cent moisture)	2	2					4
Hemlock, western (Tsuga heterophylla)							
(Oven-dry wood)	1	1	—	—	—	2	
(Wood having 13-30 per cent moisture)	1	2	1	—	1		5
Larch, western (Larix occidentalis)							
(Oven-dry wood)	2	1	—	—	—	3	
(Wood having 11-13 per cent moisture)	8	4	1				13
Maple, sugar[a] (Acer saccharum)							
(Oven-dry wood)	5	—	—	—	—	5	
(Wood having 5-28 per cent moisture)	11	9	6	1			27
Oak, red[a] (Quercus sp.)							
(Oven-dry wood)	1	3	—	1	—	5	
(Wood having 6-20 per cent moisture)	3	3	6				12
Oak, white[a] (Quercus sp.)							
(Oven-dry wood)	—	—	—	—	—		
(Wood having 5-20 per cent moisture)	1	6	11				18
Pine, southern yellow (Pinus sp.)							
(Oven-dry wood)	5	1	1			7	
(Wood having 5-24 per cent moisture)	29	26	3	1			59
Pine, white (Pinus sp.)							
(Oven-dry wood)	5	3	2			10	
(Wood having 4-18 per cent moisture)	8	8	4				20
Prima vera (Tabebuia bonnell-smithii)							
(Oven-dry wood)	1					1	
(Wood having 1-15 per cent moisture)	7	2					9
Redcedar, western (Thuja plicata)							
(Oven-dry wood)	3	—	—	—	—	3	
(Wood having 10-20 per cent moisture)	8	3	5				16
Redwood (Sequoia sempervirens)							
(Oven-dry wood)	6	1	1	—	—	8	
(Wood having 6-13 per cent moisture)	8	7					15
Spruce, Engelmann (Picea engelmannii)							
(Oven-dry wood)	3	1	—	—	—	4	
(Wood having 12-13 per cent moisture)	8	3	1				12
Tanguile (Shorea polysperma)							
(Oven-dry wood)	2	2	—	—	—	4	
(Wood having 6-15 per cent moisture)	5	1	—	—	—		6
Miscellaneous species							
(Oven-dry wood)	—	—	—	—	—		
(Wood having 7-16 per cent moisture)	5	3	4				12
Totals							
(Oven-dry wood)	48	24	9	3	—	84	
Per cent of total	57.1	28.6	10.7	3.6			
(Wood having varying amounts of moisture)	160	126	77	16	6		385
Per cent of total	41.5	32.8	20.0	4.2	1.5		

[a] If 0.12 is added to all computed values for the runs on heartwood specimens of maple, oak and greenheart the following results are obtained:

Totals							
(Oven-dry wood)	50	22	10	2		84	
Percent of total	59.5	26.2	11.9	2.4			
(Wood having varying amounts of moisture)	192	118	54	15	6		385
Per cent of total	49.9	30.7	14.0	3.9	1.5		

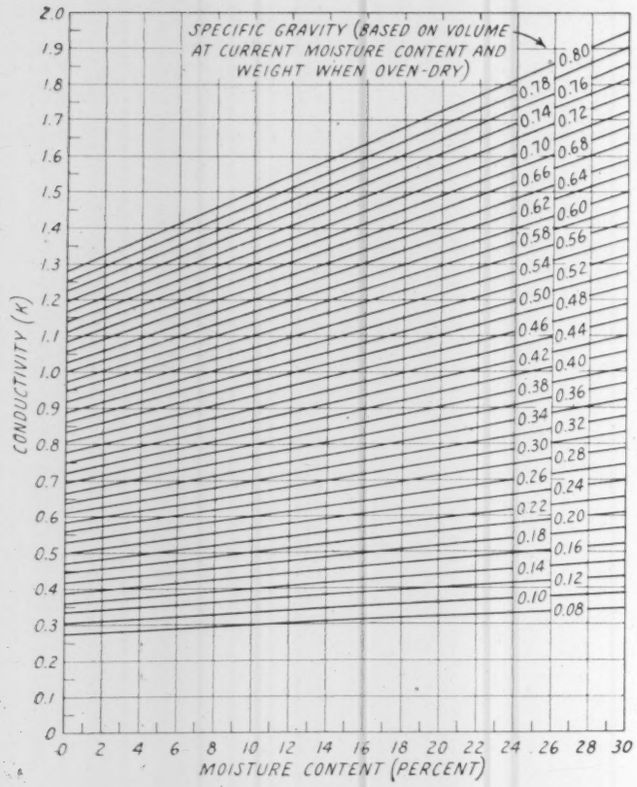
SPECIFIC GRAVITY (BASED ON VOLUME AT CURRENT MOISTURE CONTENT AND WEIGHT WHEN OVEN-DRY)
CONDUCTIVITY (K)
MOISTURE CONTENT (PERCENT)

Fig. 7—Relation between computed conductivity and moisture content for wood having different specific gravity values

(Specific gravity based on volume at current moisture content and weight when oven-dry. Conductivity computed from formula $K = S\ [1.39 + 0.028M] + 0.165$.)

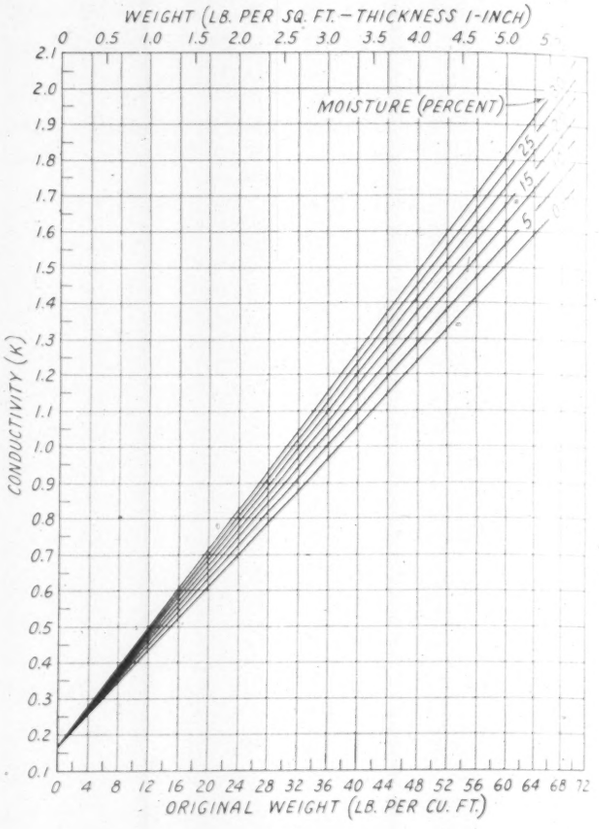

Fig. 8—Relation between computed conductivity and original weight of wood (weight at current moisture content M) with moisture content ranging from 0 to 30 per cent

Conductivity computed from formula, $K = S\ (1.39 + 0.028M) + 0.165$

$$K = \frac{Sg\ [1.39 + 0.028M]}{1 - Sg\ [0.009\ (30 - M)]} + 0.165$$

This equation was used for computing the curves in Fig. 9 which show the relation between the computed conductivity and moisture content for woods having various specific gravity values based on volume when green and weight when oven-dry. The following example illustrates the use of Fig. 9. Assume it is desired to find approximately the average conductivity of loblolly pine at 15 per cent moisture. Table 8 shows the average specific gravity of this wood when green is 0.47. By following the vertical line for 15 per cent moisture to a point halfway between the lines for specific gravities of 0.46 and 0.48 the conductivity opposite this point (read on left hand scale) is shown to be about 1.07. If we wish to compare the conductivity of this species with that of another wood, for example, yellow birch, Table 8 shows the average specific gravity (based on volume when green) of the latter species is 0.55. Fig. 9 shows that the conductivity of yellow birch at 15 per cent moisture would be about 1.24 or nearly 16 per cent more than the conductivity of loblolly pine at this moisture content. It should be noted that the specific gravity values Sg shown in Fig. 9 are based on oven-dry weight and volume when green while the specific gravity values given in Fig. 7 correspond to Sm and are based on oven-dry weight and volume at current moisture content. The latter figure is convenient to use when the specific gravity and moisture content of seasoned wood is determined or assumed.

It may be noted from Fig. 9 that loss in moisture content reduces the conductivity to a greater extent than the accompanying change to higher density increases it, hence the net result is a decrease in conductivity as wood seasons from the green to the oven-dry condition.

The small number of tests made on Douglas-fir specimens to compare conductivity in the radial and tangential directions, showed no pronounced differences although, in practically all cases, there appeared to be a tendency for higher conductivity in the tangential direction. For average wood, however, there would probably be no important difference in conductivity for these two directions. On the other hand there is a very marked difference in the conductivity in a longitudinal direction compared with that at right angles to the fibers. In several runs on Douglas-fir and red oak specimens in which the moisture content ranged from about 6 to 15 per cent, the conductivity in the longitudinal direction was from about 2¼ to 2¾ times the conductivity in the radial or tangential direction.

Experiments made in special runs to study the effect of knots, checks, and cross-grain structure indicated that small knots, when not numerous, had no important influence on conductivity but large knots had a tendency to increase the conductivity. It was also noted that there was little if any effect on conductivity when the

specimens had small checks, such as often occur in wood while seasoning. Wood with pronounced cross-grain (from face to face) showed increased conductivity, as would be expected. In wood having such cross-grain, there is a longitudinal component that increases the conductivity to a greater or less extent depending on the angle of cross-grain.

Results of the experiments on specimens of Douglas-fir plywood indicate that the thin film of glue between the wood surfaces has no important effect on conductivity. This might be expected because of the very slight thickness of the glue coating. Even a marked difference in the conductivity of the glue in comparison with the conductivity of the wood would make little difference in the conductivity through the entire thickness because the total thickness of the glue is insignificant in comparison with the total thickness of the wood.

A few runs were made to investigate the effect of differences in average temperature on the conductivity of the wood. The temperature differences between the hot and cold sides ranged from about 22 to 96 F and the differences in the average temperatures obtained in runs on a given pair of test specimens (differences in average of hot-plate plus cold-plate temperatures) varied from about 12 to 25 F. Changes in temperature were obtained by varying the temperature of the hot plate. These runs were made on both air-seasoned and oven-dry woods. In most cases, there was a small increase in conductivity with increase in average temperature, but the increase varied from nearly zero to a maximum of less than 4 per cent. In other words, for the range of hot- and cold-plate temperatures used, there was but little change in conductivity when the difference in temperature between the hot- and cold-plate sides was more than doubled by increasing the hot-plate temperature. This indicates that differences in the average temperatures of wood which are normally encountered, have no important bearing on heat conductivity.

Table 8—Average Specific Gravity of Various Species of Softwoods and Hardwoods

Species	Specific Gravity Based on Volume When Green and Weight When Oven Dry = *Sg*
Hardwoods	
Ash, white (Fraxinus americana)	0.55
Basswood, American (Tilia glabra)	0.32
Beech, American (Fagus grandifolia)	0.56
Birch, yellow (Betula lutea)	0.55
Blackgum (Nyssa sylvatica)	0.46
Chestnut, American (Castanea dentata)	0.40
Elm, American (Ulmus americana)	0.46
Elm, rock (Ulmus thomasii)	0.57
Hackberry (Celtis occidentalis)	0.49
Hickory, mockernut (Hicoria alba)	0.64
Maple, silver (Acer saccharinum)	0.44
Maple, sugar (Acer saccharum)	0.56
Oak, red (commercial) (Quercus sp.)	0.56
Oak, white (commercial) (Quercus sp.)	0.59
Pecan (Hicoria pecan)	0.60
Sweetgum (Liquidambar styraciflua)	0.44
Sycamore, American	0.46
Tupelo, water (Nyssa aquatica)	0.46
Walnut, black (Juglans nigra)	0.51
Yellowpoplar (Liriodendron tulipifera)	0.38
Softwoods	
Baldcypress (Taxodium distichum)	0.42
Douglas-fir, coast (Pseudotsuga taxifolia)	0.45
Fir, white (Abies sp.)	0.37
Hemlock, eastern (Tsuga canadensis)	0.38
Hemlock, western (Tsuga heterophylla)	0.38
Larch, western (Larix occidentalis)	0.48
Pine, eastern white (Pinus strobus)	0.34
Pine, jack (Pinus banksiana)	0.39
Pine, loblolly (Pinus taeda)	0.47
Pine, lodgepole (Pinus contorta)	0.38
Pine, longleaf (Pinus palustris)	0.54
Pine, ponderosa (Pinus ponderosa)	0.38
Pine, red (Pinus resinosa)	0.44
Pine, shortleaf (Pinus echinata)	0.46
Pine, slash (Pinus caribaea)	0.56
Pine, sugar (Pinus lambertiana)	0.35
Pine, western white (Pinus monticola)	0.36
Redcedar, eastern (Juniperus virginiana)	0.44
Redcedar, western (Thuja plicata)	0.31
Redwood (Sequoia sempervirens)	0.38
Spruce, Engelmann (Picea engelmannii)	0.31
Spruce, Sitka (Picea sitchensis)	0.37
Spruce, white (Picea glauca)	0.37
Tamarack (Larix laricina)	0.49
White-cedar, northern (Thuja occidentalis)	0.29

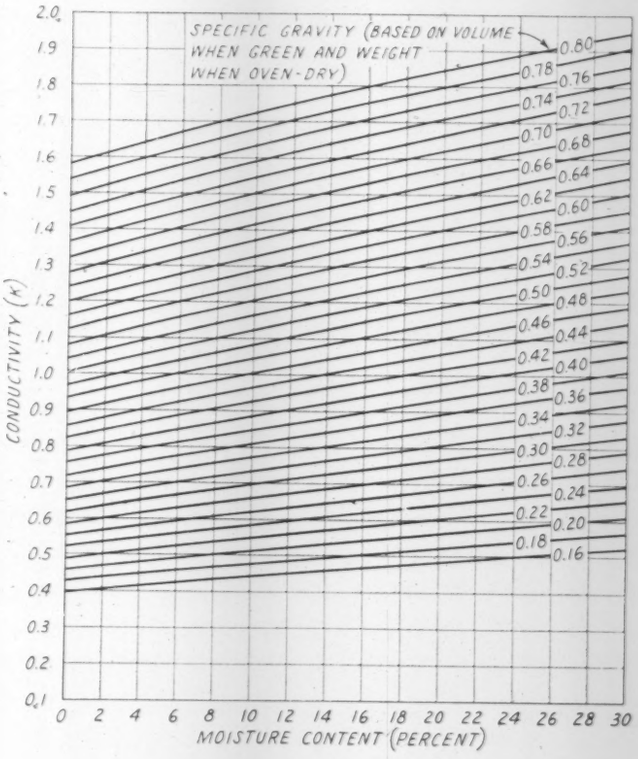

Fig. 9—Relation between computed conductivity and moisture content for woods having various specific gravity values based on volume when green and weight when oven-dry

The Mechanical Equipment of Air Raid Shelters†

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THE bombardment of cities and the use of poisonous gases in modern warfare has made the providing of air raid shelters necessary to protect and save human life. Such shelters may be simply a cellar of a home, sometimes with reinforced ceiling constructions, or specially built shelters housing large numbers of people. Underground vaults and passages, such as subways, are often used. Larger shelters not only present special construction problems but also present problems to the engineer in creating proper atmospheric conditions for maintaining health and providing equipment to carry on the necessary and normal activities of life.

The construction of air raid shelters in France, England, Germany and Switzerland has shown that the necessary mechanical equipment not only requires special study, but that in larger and more important shelters the cost of such equipment must be secondary to safety. A comparison of the cost of such equipment with the cost of the shelter itself shows that quite often the first greatly exceeds that of the latter. But it is often possible with more complete or additional mechanical equipment to reduce the building costs, as for example, with mechanical ventilation the same number of people can be housed in a smaller shelter thereby reducing the initial building costs.

The construction of air raid shelters presents the mechanical engineer with the following problems: 1. That in a given space, which through special building construction has been made more or less bomb proof, a larger number of people can find shelter for long periods of time without detriment to their health. 2. The possibility that the providing of necessary services, such as water, electricity, etc. from outside of the shelter may be interrupted or entirely destroyed. 3. That protected rooms be provided for maintaining or providing necessities, such as water, electricity, and telephone and for the housing of police, firemen, air raid personnel, etc.

To solve these problems the following mechanical equipment is necessary:

1. Ventilating systems.
2. Independent production of light and power.
3. Auxiliaries as water pumps, sewage pumps, signal apparatus, telephone, etc.

Ventilating Systems

Each person requires for the maintenance of life about 70 cu ft of air per hour and gives off about 0.88 cu ft of carbon dioxide. While life can be prolonged in a closed room without ventilating by reducing the carbon dioxide content, the necessity of ventilating enclosed spaces containing many persons is quite evident.

Many trials, especially in fortifications and air raid shelters already constructed, have shown that an absolutely air tight room can be built only out of metal. With masonry or similarly constructed air raid shelters, in addition to the air leaks around doors and windows, there is also always the possibility of air leakage through the structure itself. Even in structures which appear to be air tight, gases will enter the shelter from outside with the slightest outdoor air pressure. Therefore the importance of keeping the air within the shelter under a positive air pressure to keep poisonous gases out is quite evident and necessary.

One method of maintaining a positive air pressure in the shelter is by the use of compressed air bottles. Such equipment requires very careful building construction for air tightness, otherwise too much compressed air would have to be used. The other method, which is the better one to use, is to obtain the required positive air pressure in the shelter through the introduction of gas filtered air taken from outdoors and forced into the shelter by means of fans. Such ventilation is necessary in every shelter where many people assemble and remain for a long period of time.

Special attention must also be given to the heat and moisture given off by occupants of the shelter. This for active persons amounts to 600 Btu and 2530 grains of moisture per hour per person. Under these assumptions, a shelter containing 100 people would produce 60,000 Btu per hour and 36.2 lb of moisture. The relatively cold walls, floors and ceilings of the air shelter will naturally absorb some of this heat. The minimum required amount of air to be introduced into the shelter per person every hour varies in the different European countries for example: Switzerland requires 106 cfh; Germany 53 cfh; and, France 70 cfh. These limited air quantities, while probably sufficient for short periods of occupancy in the shelter, would not be sufficient for longer periods as sometimes the occupants must stay in the shelter 8 hours or more at one time. The dew-point is generally quickly reached in an air raid shelter due to the cooling off of the air by the shelter walls, floors and ceilings and the addition of moisture by the occupants themselves. This is the reason that, even with European climatic conditions, shelters must not only be ventilated, but in many cases these shelters should also be provided with cooling and humidity control. The desired air conditioning could probably be obtained through the use of a liberal and large amount of air but this would require considerable electric power and larger equipment, all of which should be avoided in air raid shelter design. That is the reason why complete ventilating systems are installed, not only for air movement and gas filtering, but also for heating by means of electricity or warm water and for cooling with

†Translation from the German by Alfred J. Offner, Consulting Engineer, New York, N. Y.
*Director of LUWA Ltd.

street water supply, well water or by mechanical refrigeration, the latter of which requires necessary additional electric and power equipment provided independent of outside sources.

Air raid shelters may be ventilated in two ways: (a). By an air supply and air exhaust ventilating system. The air is taken in from outdoors, either unfiltered or gas filtered and then blown into the shelter by a blower, putting the room or rooms under pressure. Part of the air will escape from the building due to leakage through the structure itself and the remaining through specially designed excess pressure valves. (b). By complete recirculation of the air, without the introduction of any outdoor air, the recirculated air being regenerated by absorption of the carbon dioxide and the addition of oxygen. In this case, to maintain the positive pressure required in the room for safety, a certain additional amount of compressed air, generally provided by a bottle system, will be required.

The following is a description of the special parts required and the precautions that have to be taken with such ventilating installations. As it is possible to have an outdoor air intake blocked or destroyed by an outdoor explosion, it is always advisable to have several outdoor air intakes, located in different parts and sides of the building. If two outdoor openings are provided, one should be used for ventilation without gas filtration and the other for use during poisonous gas attacks. Such method of operation keeps one air intake clean of gas contamination. An air intake, through which poisonous gases have passed will be unclean and always remain unusable for a long time for ventilation without the use of the gas filtering units. Following this precaution will avoid the possibility of having the air in the shelter poisoned by gases that may remain in the inlet duct. In addition to the clean air inlet to be located near the level of the street, if possible, another inlet should be provided, preferably in a location high above the street level. While old chimneys or similar masonry shafts can be used for such purposes if they are in good condition, it is better to use air passages built out of metal.

Immediately at the point where the outdoor air enters the shelter, there should be provided an air-tight hand-operated damper, similar to type as illustrated in Fig. 1. These dampers must be of heavy construction to protect them against possible damage by explosions outside of the air intake. Such dampers must be operated quickly and should be so arranged that they can be opened or closed with one movement of the hand, and locked in position by a brake arrangement similar to that used in automobiles. A similar hand-operated damper should be installed in the by-pass duct connection provided around the gas filter. Outdoor and conditioned air is distributed throughout the room or rooms with duct work so arranged to give positive and thorough ventilation to all parts of the structure.

The ducts are generally constructed of galvanized iron and are located at the ceilings of the rooms as shown in Fig. 2. The air supply inlets should be adjustable for air direction and air quantity control and so arranged that the adjustments cannot be changed by the occupants.

As already explained, the exhaust air will partly leave the shelter by air leakage through the building construction and the greater amount through the excess pressure valves provided for direct air discharge to the outdoors. The building walls are calculated for excess pressure of from 5 to 10 millimeters water column. Should the ventilation system break down, or in case of failure of the outdoor air intake, making air recirculating necessary, these outlet valves must be closed air tight from the inside. Air tight excess pressure valves of all metal construction are preferable to those using water for a seal as at a critical moment there is a possibility of failure of the water seal due to lack of water or freezing.

Ventilators and Their Operating Power

Due to the relatively high air pressure drop caused by the gas filters it is preferable to use fans of the blower type. The gas filters should be on the suction side of the blower so that the air is drawn through the filters. The reason for this is, should the outdoor air intake either leak or be contaminated, it is impossible for poisonous gases to enter the shelter before first being filtered. The fan then blows the filtered air

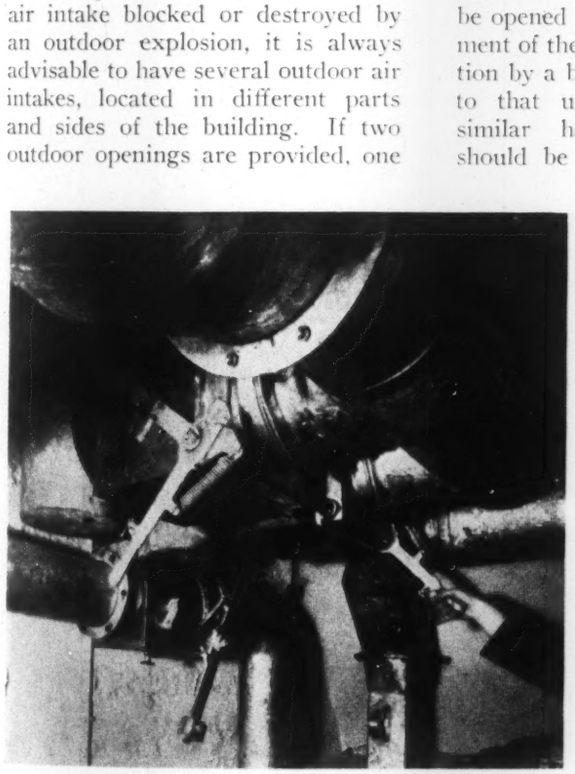

Fig. 1—Outdoor air intakes showing special type of air tight dampers

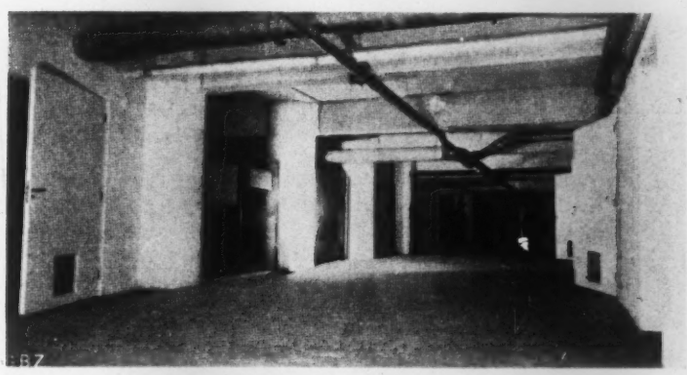

Fig. 2—The duct system for a shelter in a bank. The ducts have to be designed so that they may be removed and thoroughly cleaned after their contact with poisonous gases

through necessary air heaters, air coolers and dehumidifying apparatus. A by-pass duct connection provided with a damper should be installed around the gas filter so that it is possible to ventilate the shelter without using the gas filters during periods when there is no gas attack, thereby saving wear and maintenance of the gas filters.

For normal operation of the blowers electric motors are generally used. Means for turning the blower by man power, in case of electric current failure, should be provided. Such manually operated equipment is recommended even if the shelter is provided with its own electric generating plant as the latter may fail in an emergency.

For smaller installations, up to a maximum of 60 w, a hand crank can be used. The electric motor should be so arranged that it can be disconnected from the shaft making it unnecessary to also turn it by hand.

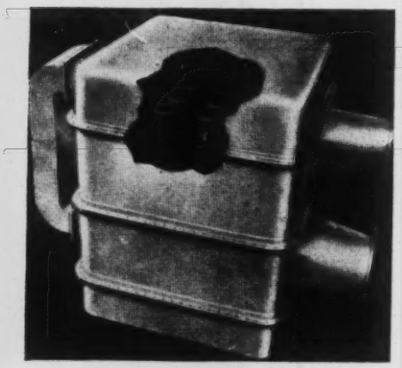

Fig. 3—Standard element gas filter showing position of single elements. According to the quantity of air, more or less of these elements are necessary

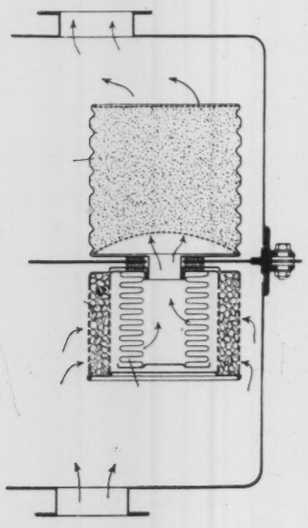

Fig. 4—Cross-section of a gas filtering element

For larger blowers, which cannot be turned by hand, a foot pedal drive similar to that used on a bicycle should be provided. With pedal operation it is also possible to attach a small dynamo for emergency lighting, at least enough to illuminate the room containing the equipment. A simple air quantity indicator should be provided on the blower discharge so that the operator of the hand crank or foot pedal would know the speed at which he has to operate the fan to give the desired air quantities.

Poisonous Gas Filters

The poisonous gases used in modern chemical warfare may be divided into two types, (1) certain true gases, in the chemical sense, as for example phosgene, chlorine, etc. (2) clouds or smoke of suspended semi-solid particles, the so-called *Poison Smoke* or *Arsin* containing arsinical or other compounds.

These poisonous gases are discharged by the enemy by various methods, as shells, bombs and pressure containers, all for the injury to or destruction of human life.

In general, there are two different principles of filter construction, one is the block type and the other the standard element type. As the name implies, the block type is constructed in a single container and is effective in filtering out the arsine as well as the gas. With the block type of filter there is always present a possible danger, as should the density of the carbon fill not be uniform over the entire filtering surface, some gases may pass through the weak spots too quickly and consequently not be entirely eliminated. This danger is increased if the filters have to be transported over great distances as in transit they might be jarred. The resistance to the flow of air in the block type filter is higher than for the standard element type, while the cost of the former is less than the latter.

The standard element type of filter is illustrated in Fig. 3. It is composed of a large number of independent elements, each element being rated for a fixed quantity of air, generally 530 cu ft per hour of air, so for example 20 elements would be required to filter 10,600 cu ft of air per hour. In Fig. 4 is shown a cross-section of one of these elements. The air to be filtered enters the element from the sides. The air first passes through a prefilter to take out the larger dust particles so as to protect to a certain degree the following arsin filter which is the most important part of the filtering system. The arsin filter is composed of a special filtering material which has an appearance similar to blotting paper. This material is folded in a circular accordion fashion and gives the greatest filtering area with the least air resistance. The air passes through the center of the filter and then flows over to the gas filter. This gas filter is a small separate container and each element filters only a small quantity of air. The activated carbon in the gas filter is placed into these small elements

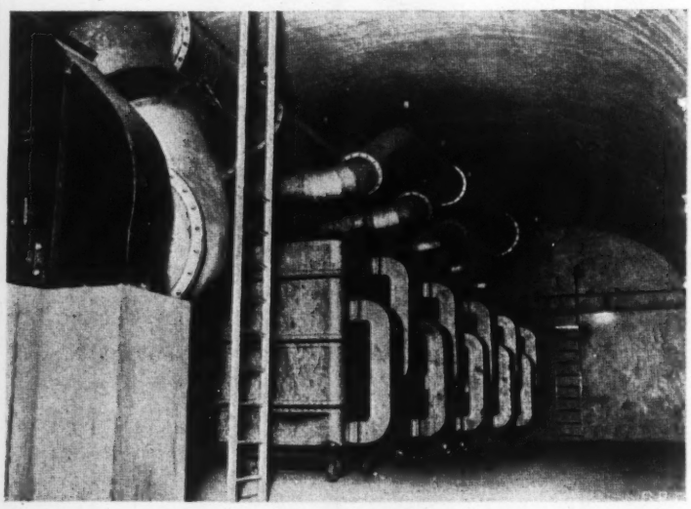

Fig. 5—Gas filtering system located in a Paris subway station. There are five filter containers, each container filtering sufficient air for 1,000 people

very compactly and is pressed down by a spring so it cannot move, thereby preventing break-throughs of the air. The arsin filter and the carbon filter are screwed together as one unit.

The total air flow resistance of the entire element type of filter is only 50 millimeters water column, while the block type filters generally have an air flow resistance of 150 to 250 millimeters. The greater resistance of the block type filter not only increases the power necessary to drive the ventilating equipment, but also is objectional when the equipment has to be turned by man power. Another advantage of the standard element type filter is the facility with which the elements can be exchanged, as all elements are standard and it is unnecessary to have exchange elements in each shelter. This type of gas filter has been adopted in England, France and Switzerland and now is also being used extensively in Germany where previously the block type of filter was used exclusively. A group of gas filters for a large shelter is shown in Fig. 5.

Absorption of Carbon Dioxide

The air to be circulated in a complete recirculating system must be regenerated by the absorption of carbon dioxide in specially designed devices which have been developed for this purpose. Two such units, which are metal cases filled with soda and lime, can be seen in the background of Fig. 6, the air to be regenerated being drawn through these devices from below. In smaller systems the blower used in the ventilating system is also used for the absorbers, while in larger installations a second blower might be necessary for the absorbers. As the air passes up through the unit the lower portion of the soda-lime is used up first. The device is so constructed that one single movement releases the used chemicals from the box below and one has simply to refill the device from the top. During the process of carbon dioxide absorption heat is being developed, and to control this absorbing process two thermometers are attached to the absorber and the rate of flow of air adjusted so that the temperature changes are kept within limits. The regenerated air is then distributed throughout the shelter by means of the ventilating duct system. With the absorption of carbon dioxide a partial negative pressure is created in the shelter with the ensuing danger of poisonous gases entering through building structure leakage. To counter-balance this possible danger compressed air or oxygen from bottles must be added. The use of oxygen is preferable as a make up for that used by the occupants of the shelter. There are other methods of absorbing carbon dioxide as with alkali peroxides, which at the same time also release oxygen. The disadvantage of this latter described system is that it needs special sources of heat and humidification of the air. Its application is relatively complicated and the necessary chemicals are expensive. The use of the soda-lime type of system is preferable as the system is cheaper and the chemicals are available in large quantities.

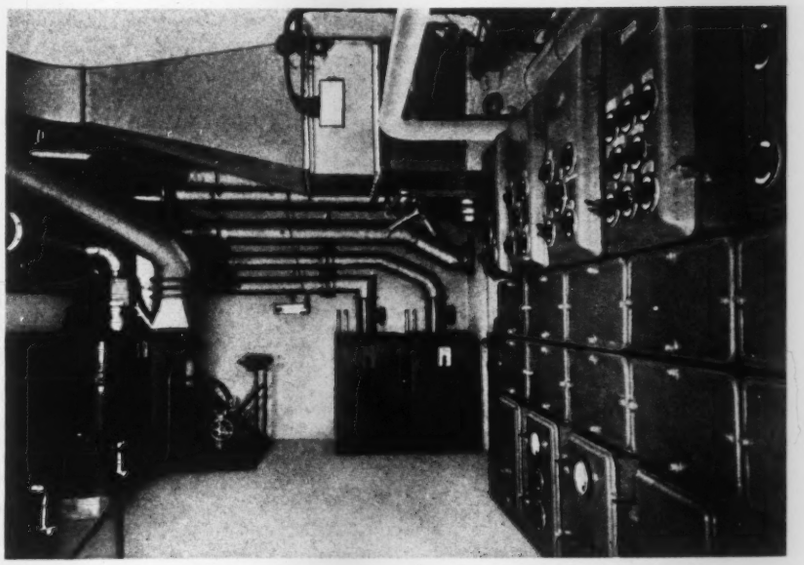

Fig. 6—Mechanical equipment of a 300 person shelter located in Paris. To left are shown standard element gas filters and four bicycles for ventilating purposes. In the background are two carbon dioxide absorbers and on ceiling are ducts for air conditioning system. On right is the switchboard for the emergency Diesel generator and compressors

For the addition of oxygen, the standard oxygen bottles available to the trade are used. Experience has proven the necessity of buying these oxygen bottles in peace time as with the start of a war, there is so great a demand for them, that few bottles are available for civilian use. The oxygen at a low pressure flows through a nozzle into the suction side of the blower. There should be a scale indicating the exact amount of oxygen discharged, and the correct amount should be regulated by the person in charge of the equipment so that not more than 30 liters of oxygen per person are supplied per hour. The adverse influence of having too much oxygen in the air is generally known. As the possibilities of errors with hand regulation of the oxygen supply are many, it is advisable to install a meter similar to the standard gas meter to control the quantity of oxygen released in a given time. In this case, the scale should be so arranged that the person in charge of the shelter can easily check the output without having to make calculations. An oxygen meter as well as the carbon dioxide absorber and the manometer are shown in Figs. 7 and 8.

Heating

Any heat required for the shelter should not be supplied from an outside source, due to the possibility of such service being interrupted or destroyed. As the occupants of the shelter give off considerable heat, the incoming air need not be heated to a temperature higher than 68 F, which air temperature is even warm enough with very low outdoor temperature. It is important to have the incoming air tempered, as experience has shown that the ventilating system cannot be used during the winter time if the outdoor air is not preheated. In several shelters where no provision for tempering the air

was provided, heating units had to be installed after the shelters were finished.

The heating of shelters used for emergency hospital purposes requires more careful study and additional electrical heaters and equipment should be installed to supplement the heating provided by the ventilating system. In designing this additional equipment special attention must be given to the length of time required to heat up the shelter which should be relatively short. In some of these special shelters a second system of ducts has been used for recirculating part of the air, which is further heated, and such systems have proven efficient.

Cooling and Drying

In Switzerland, as already stated, 106 cfh of air per person is considered a normal quantity of air to be introduced into a shelter, which amount is just enough to keep the occupants in a fairly comfortable condition. It is evident, however, that this amount of air would not be sufficient to remove the heat and humidity produced by a person and for this reason if this amount of air is used, the shelter would quickly become saturated, and the air temperature rise rapidly. Therefore the cooling of shelters would seem an absolute necessity. To take care of this condition the quantity of air must be increased from the 106 cfh minimum to between 500 and 900 cfh per person. This increased amount of air necessitates larger ventilating systems and equipment and increases the installation costs so greatly that in Europe only shelters where people do active work have been air conditioned. Some shelters have been dehumidified by silica gel units. The disadvantage of such systems is that while dehumidification is provided for, no cooling of the air is effected, which in America is probably the most important problem. An air conditioning unit is shown on the right-hand side of Fig. 2 and a cooling coil is shown on the ceiling in Fig. 8. In this latter case the cooling is being done by water from the city water mains.

Emergency Power Equipment

As a protection against possible breakdown or destruction of the outside electricity service supply, important shelters have to be provided with independent emergency electric generating plants. As breakdowns in the outside electrical supply systems might occur frequently, it is advisable to select emergency plants larger in capacity than required, so that a part of the electricity generated can, in case of an emergency, be supplied to parts of the building outside of the shelter.

For the installation of emergency electric generators the following items must be taken care of:

(1) The generators, whether Diesel engine or gasoline driven, require large quantities of air and this required air should also be gas filtered. A separate gas filter and blower should be provided for the emergency generator. A common air inlet can be used for both the ventilating system and the emergency plant. The emergency plant blower need not be arranged for operation by man power as such

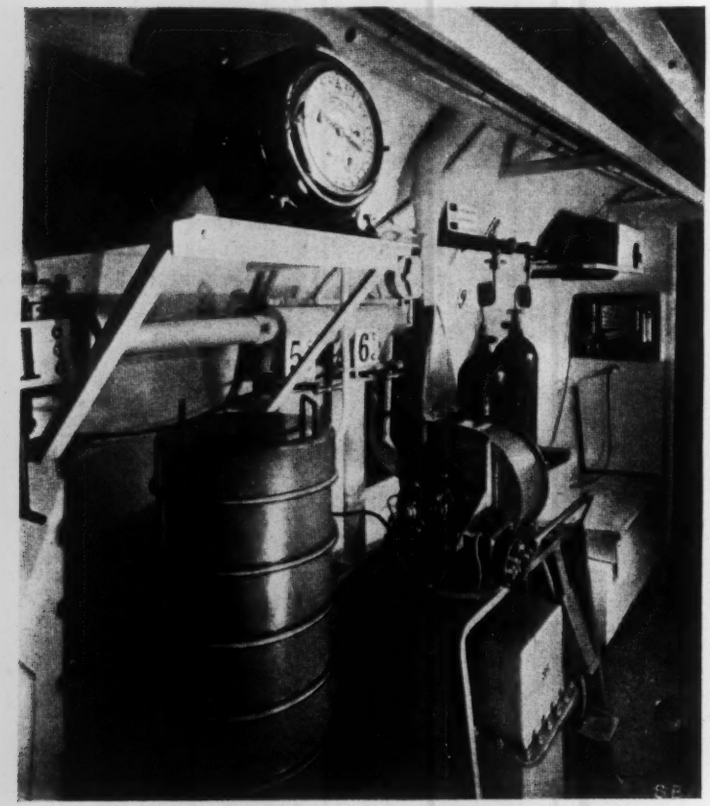

Fig. 7—Equipment in a 20 person private shelter located in England. At left is a carbon dioxide absorber. Above, a meter to control oxygen output. At right, a standard element gas filter. In the background, oxygen bottles, pressure control and gas detector. On the ceiling, cooling coils for air conditioning the shelter

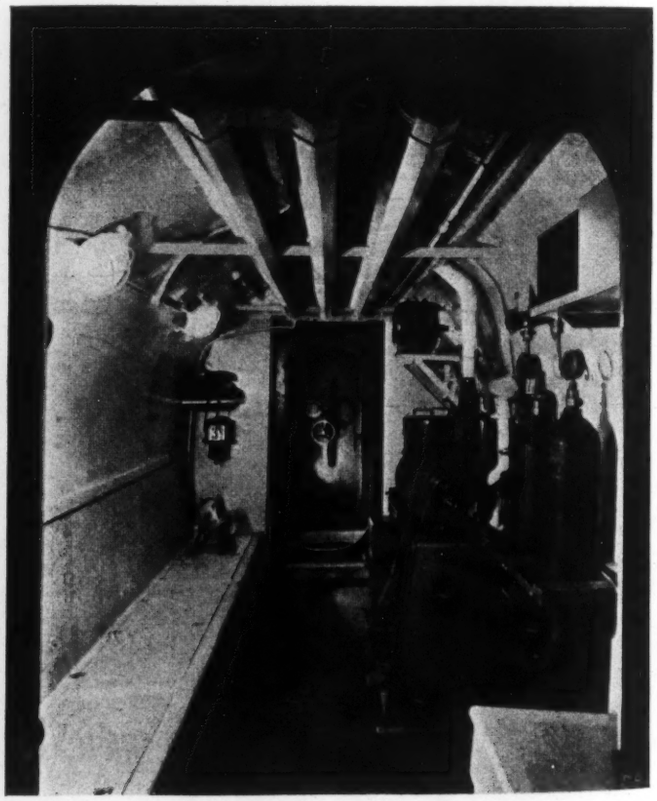

Fig. 8—Same shelter as Fig. 7 seen from other side. In the background is entrance lock to shelter

blowers need only operate when electricity is being generated. As the generator produces large quantities of heat it is necessary to provide air for cooling. On account of the large costs involved in gas filtering the air required for the emergency generator, many shelters have used the air coming directly from outdoors without gas filtering, while in several cases the excess air in the shelter has been used. The emergency generating plant should be installed in an air tight room as part of the shelter itself and have the same protection as provided for the other parts of the shelter. A small door is generally provided for communication between the two rooms. This separation is a necessity for two reasons; first the noise of the generator is considerable and is apt to upset the occupants of the shelter and make them more nervous and secondly, it eliminates the possible danger of carbon monoxide gases escaping into the shelter.

(2) All parts of the generating plant must be insulated against vibration and noise. Special care must be taken in the construction of the exhaust pipe and its discharge to outdoors must be as far away as possible from the ventilating system's outdoor air intake. The same precaution as taken in other parts of the shelter must be taken for the removal of the excess air in the emergency generator room, and heavy excess pressure valves must be provided, controllable from the inside of the room.

A Diesel generating unit is preferable to a gasoline driven unit even in spite of the higher cost of the former. A gasoline unit located in a small room is a fire hazard. There is also the difficult problem of the gasoline storage and the evacuation of the gasoline vapors from the engine. Smaller Diesel units up to 12 kilowatt capacity can be started by hand. Larger units require an automatic starting device with compressed air. All this adds to the installation costs. The usual Diesel engine requires water for cooling for which purpose city water can be used. To be independent of this outside water supply requires either the installation of separate water tanks large enough to provide cooling water for several hours or the digging of wells.

As fuel delivery in wartime is not dependable, it is evident that a sufficient amount of fuel must be stored at one time to provide for a long period of operation. If the fuel is delivered to the shelter in barrels, a hand pump must be provided to transfer it into the storage tank. In addition to a large storage tank, it is advisable for safety in operation also to install a small or day tank containing sufficient fuel for one day's operation.

Accessory Equipment

In large shelters where the recirculation system of ventilation is used a special air compressor must be installed to provide compressed air for filling the steel bottles. As the air compressor is generally used only after an attack, the load of the compressor motor need not be added to the load on the emergency generating plant.

In cases where water is to be provided independent of outside sources, well water with a pump is used and this water is stored in pressure tanks using compressed air. Such equipment might be necessary in shelters several stories in height, especially for toilet use or for air conditioning systems.

As it is desirable to have air raid shelters deep in the ground, a special pump has to be provided for sewage disposal from the sanitary system and this sewage is pumped into the regular community sewage system. These pipes have to be protected by siphons large enough to prevent the infiltration of sewage gases into the shelter.

For shelters used as commander posts and telephone terminals special accumulator batteries are required. These accumulators can be combined with the emergency generator plant or a special transformer has to be installed. In either case no outside electricity supply should be used to start these accumulators. It must be remembered that the gases of these batteries are poisonous and must be removed by a separate exhaust ventilating system.

Operation

The person in charge of the shelter and of the equipment must be fully informed and instructed as to the care, operation and purpose of the entire plant. It is also very important that framed typewritten operating instructions be hung in each shelter and these instructions should be short and carefully written. All control handles, valves, switches, dampers, etc., and all apparatus and equipment should be numbered and have their purpose, control, operation and name clearly stated in large letters. It must be remembered that, in case of an air raid alarm, confusion and nervousness always occur.

CORROSION TOPIC AT WESTERN MICHIGAN

March 10, 1941. The March meeting of the Western Michigan Chapter was held at the Burdick Hotel, Kalamazoo, with 62 members and guests in attendance.

Following the report by the treasurer, W. L. Warnock commented on the program for the annual meeting and stated that it would be held late in May in Grand Rapids with the exact date and place to be announced later.

The speaker of the evening, L. F. Collins, was then introduced and an instructive talk, which he illustrated by slides, was given on corrosion in steam heating systems. Special comment was made to the effect that the corrosion problem will have to be solved by the heating and the chemical engineers working together, and that neither of these would be able to do a satisfactory job alone. It was also pointed out that ample air venting tended to decrease the amount of oxygen and CO_2 which are the gases doing most of the damage.

PHILADELPHIA EXPECTS

You In '42 . . . .

Only 7 Months Until

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MEETING

January 26-30, 1942

PROGRAM

Semi-Annual Meeting 1941

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Palace Hotel — **June 16-20, 1941** — **San Francisco, Calif.**

(All Events on Pacific Standard Time)

Sunday, June 15

10:00 a. m. REGISTRATION—Palace Hotel (*Concert Room—Lobby Floor*)

2:00 p. m. Committee Meetings—(See Bulletin Board at Registration Desk)

Monday, June 16

8:30 a. m. REGISTRATION—Palace Hotel (*Concert Room—Lobby Floor*)

10:00 a. m. JOINT SESSION with *Heating, Piping and Air Conditioning Contractors National Association*—St. Francis Hotel (*Colonial Room*) W. L. Fleisher presiding

(*a*) Invocation
(*b*) Singing "America"
(*c*) Greetings from San Francisco
(*d*) Response by J. E. McNevin, President, *HPACCNA*
(*e*) Greetings by Clyde E. Bentley, President, Golden Gate Chapter, and Daniel Hayes, President, *Heating and Piping Contractors Association of San Francisco*
(*f*) Program Announcements
(*g*) Group photograph
(*h*) Address: A. Appleton, Pres. San Francisco Chapter *AIA*

12:15 p. m. Joint Luncheon—Hotel St. Francis—Tickets $1.25 per person—sponsored by *HPACCNA*

2:00 p. m. Opening of Pacific Heating and Air Conditioning Exposition—Civic Auditorium

2:00 p. m. Golf Tournament, Lakeside Country Club

2:30 p. m. Council Meeting

7:00 p. m. Dinner and Dance—Hotel St. Francis (Mural Room—Tickets $3.50 per person—Sponsored by *HPACCNA*) (ASHVE Members Invited)

Tuesday, June 17

10:00 a. m. Nominating Committee Meeting

10:30 a. m. Ladies sightseeing trip via Golden Gate Bridge to Marin County, Muir Woods (Redwoods) and Mt. Tamalpais (Luncheon $1.00 at Mt. Tamalpais Tavern—Elevation 1500 ft—Transportation free) Buses will return to Palace Hotel at 4:00 p. m.

1:30 p. m. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)

Amendments to By-Laws
Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson
Weather Data for Cooling Design Application, by J. D. Kroeker
Concerning Conservation of Underground Water with Suggestions for Control, by Noel E. Porter
Recent Developments in Absorption Refrigeration, by Glen Miller

12:00 p. m. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

7:00 p. m. Sightseeing (as selected by individual members; no planned Entertainment)

Wednesday, June 18

9:30 a. m. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)

The Application of Gas Fuel to Existing Heating Boilers, by R. L. Grutzmacher
The Thermal Conductivity of Wood, by J. D. MacLean
Effect of Insulation on Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo
An Analysis of Factors Influencing Building Heat Losses, by Paul D. Close

10:00 a. m. Ladies assemble for scenic ride to the interesting places of San Francisco—last Bus will leave Palace Hotel at 10:30 a. m. and return at 3:00 p. m. (Luncheon $1.00 in *Sequoia Room* of famous Cliff House overlooking Pacific Ocean and Seal Rocks—Transportation free)

12:00 p. m. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

7:00 p. m. Semi-Annual Banquet and Dance—Palace Hotel (*Palm Court*) Toastmaster, Dr. B. M. Woods, University of California—Presentation of Past President's Memory Book to Dr. F. E. Giesecke; "Human Engineering" by Dr. C. D. Leake; (*HPACCNA* members invited)—Tickets at Registration Desk $3.50 per person

Thursday, June 19

9:30 a. m. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)

Resumé of Society Research in Cooperating Institutions, by A. E. Stacey, Jr.
The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates
Economical Air Velocities for Mechanical Air Filtration, by F. B. Rowley and R. C. Jordan
Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet

11:30 a. m. Adjournment

11:30 a. m. University of California luncheon-inspection trip via San Francisco-Oakland Bay Bridge—Buses to Claremont Hotel, Berkeley, assemble at New Montgomery entrance Palace Hotel

12:00 p. m. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

12:15 p. m. Luncheon at Claremont Hotel (*Spanish Room*—Transportation and Luncheon tickets $1.75 per person; luncheon ticket only .85c)

1:30 p. m. Buses leave Claremont Hotel for University Campus, entrance via Sather Gate

2:00 p. m. Ladies sightseeing trip through University of California Campus and Buildings

2:00 p. m. TECHNICAL SESSION—University of California, Engineering Building (*Room* 104)

Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by Howard

H. Niederman, E. D. Howe, John P. Longwell, Ralph A. Seban and L. M. K. Boelter

3:00 p. m. Inspection trip for men through cyclotron and engineering laboratories

4:00 p. m. Buses will leave Campus for return trip to San Francisco as soon as loaded (last Bus to leave International House after Tea)

4:30 p. m. Tea for members and ladies at International House, University of California

Friday, June 20

ASHVE—JOINT PROGRAM—*HPACCNA*

10:00 a. m. Sightseeing and Inspection Trips

10:00 a. m. Ladies shopping tour in San Francisco downtown district—See famous "Gumps Store" and novelty shops in Chinatown (all within walking distance from Palace Hotel)

10:00 a. m. Golf Tournament (ASHVE *vs. HPACCNA*—Leave Palace Hotel by Taxi for Lakeside Country Club (cost $1.00, maximum for 4 persons—Greens fees $2.00—Assemble at Lakeside Locker Room)

12:00 p. m. to 10:30 p. m. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

2:00 p. m. Inspection trip for men and women to Pacific Gas and Electric Co., Station "A"—Transportation by Buses

Committee on Arrangements

G. M. Simonson, General Chairman; Clyde E. Bentley, Vice-Chairman; B. M. Woods, Vice-Chairman.

RECEPTION COMMITTEE: Jared A. Hill, Chairman; George G. Harrison, R. B. Holland, G. C. Cummings, George K. Brokaw

TRANSPORTATION COMMITTEE: R. A. Folsom, Chairman; Wm. E. Gee, Jr., R. A. Parker

ENTERTAINMENT COMMITTEE: Clarence L. Peterson, Chairman; Frank W. Hook, C. D. Williams, Clark E. Wayland

BANQUET COMMITTEE: Neil H. Peterson, Chairman; T. A. Marshall, H. S. Haley

LADIES COMMITTEE: J. F. Kooistra, Chairman; Mrs. J. F. Kooistra, T. J. White, Mrs. T. J. White, Mason Emanuels, Mrs. Mason Emanuels

PUBLICITY COMMITTEE: E. W. Simons, Chairman; O. W. Johnson, Edward J. Rosen

FINANCE COMMITTEE: A. J. Bouey, Chairman; L. H. Cochran, James Krueger

SPORTS COMMITTEE: F. W. Kolb, Chairman; W. P. Scott, Jr., James Gavner

COMMITTEE ALTERNATES: E. C. Cooley, Harold E. Newman, E. J. Ploskey, William W. Gee, Jr.

Pacific Coast Bids ASHVE Welcome

THE Golden Gate Chapter is ready to welcome the members and guests arriving in San Francisco for the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, June 16-20, 1941, and all plans have been completed to make this a successful meeting. The Palace Hotel will be headquarters, and the "welcome" mat is out.

On Monday the Society will hold a Joint Session with the *Heating, Piping and Air Conditioning Contractors National Association,* which is holding their 52nd Annual Convention with headquarters at the Hotel St. Francis. Pres. W. L. Fleisher will preside at this Joint Session at the St. Francis Hotel and Clyde E. Bentley, President of Golden Gate Chapter of the Society, will greet the members and guests, while Daniel Hayes, President of the local Contractors group will represent the Heating and Piping Contractors.

Following this session there will be a luncheon for both the Society members and friends and the Contractors group and guests, and in the afternoon they will attend the opening ceremonies of the Pacific Heating and Air Conditioning Exposition at the Civic Auditorium, the arrangements for which have been made by Charles F. Roth, New York.

Numerous sightseeing trips, luncheons, teas and shopping tours have been arranged to take care of the ladies, who will have little time to themselves if they wish to take part in all the activities scheduled for them. They will enjoy lunching on Mt. Tamalpais, 1500 ft elevation, planned as part of the sightseeing trip on Tuesday, and on Wednesday they will have an opportunity to enjoy luncheon in the famous Cliff House overlooking the Pacific Ocean and Seal Rocks.

While the ladies are being entertained, the members and guests will have four Technical Sessions to attend and will have an opportunity to take part in discussions on such subjects as panel heating and cooling, weather data for cooling design, conservation of underground water, use of gas fuel in heating boilers, conductivity of wood, effect of insulation on Research Residence performance, building heat losses, the interaction constant for moist air, local cooling of workers in industry, and the performance of a forced draft cooling tower.

The Semi-Annual Banquet and dance is scheduled for Wednesday evening, and G. M. Simonson, General Chairman of Arrangements, has announced that all plans have been completed for the banquet in Palm Court, at which Dr. B. M. Woods, University of California, will act as toastmaster. Armand Girard will be master of ceremonies and music will be furnished by Larry Cannon's orchestra. During the banquet, Dr. F. E. Giesecke will be presented with the Past President's Memory Book, and Dr. C. D. Leake will speak on "Human Engineering."

A. E. Stacey, Jr., Chairman of the Committee on Research, will deliver a Resumé of Society Research in Cooperating Institutions at the Thursday morning session. Mr. Stacey's talk will cover a brief digest of current research investigations illustrated with descriptive slides showing apparatus and test equipment of the various schools.

An interesting session has been arranged for Thursday afternoon at the University of California, with an inspection trip through the cyclotron and engineering laboratories.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting.

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

Exposition

MANUFACTURERS of heating and air conditioning equipment from all parts of the United States have enrolled in the Pacific Heating and Air Conditioning Exposition, June 16-20, in the Exposition Auditorium, San Francisco. Sponsorship by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in connection with its first Pacific Coast meeting, supports the prospect of a professional attendance of leading scientists, technologists and engineers also national in scope, as well as a substantial regional attendance.

Included among the exhibitors are leading producers of practically every type of appliance known to the air conditioning art in the adjoining fields of air conditioning for human comfort and air conditioning for industry.

This list gives assurance that the Exposition will afford a valuable and comprehensive review of the latest developments in the field at a time when the western region is in a most receptive situation. The West Coast territory is now undergoing the greatest industrial expansion it has ever known, with industrial progress being dramatized by the inauguration and completion of power projects, public works, airplane factories, shipyards, munitions plants, multiple dwelling operations, and the great stimulation of retailing and service to the consumer induced by shifts of population, increased employment, and enlargement of purchasing power.

The display is under the management of the International Exposition Company, of New York, which has organized and operated all of the International Heating and Ventilating Expositions held since 1930. It has therefore been assembled on a national scale, but with special emphasis on the types of equipment specially suited to the locality. Charles F. Roth is manager, and E. K. Stevens associate manager of the Exposition.

Audience appeal has been directed principally to the 11 western states by a widespread distribution of invitations to executives, superintendents and operating engineers associated with industrial, commercial and institutional enterprises, housing projects and private dwelling construction. Also included among those invited are investors, owners, supervisors, administrators and others classed as potential buyers and users of heating and air conditioning equipment.

Also qualified by vocational interest to receive invitations to the Exposition are architects and builders, and others associated with construction materials and building supplies. An important contingent of visitors is further assured by the 52nd Annual Convention of the *Heating, Piping and Air Conditioning Contractors National Association*, during the period of the Exposition.

Although the heating and ventilating expositions generally are closed to the public, in order to avoid overcrowding by mere curiosity seekers, provision has been made at the Pacific Heating and Air Conditioning Exposition for admission of persons representing mercantile and service concerns which deal with the public and therefore have a direct interest in viewing the scope of the art, studying its possible applications and meeting and consulting with representatives of manufacturers as well as engineers and contractors.

The Advisory Committee of the Exposition is composed of the following:

E. O. Eastwood, first vice president, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and head of the Department of Mechanical Engineering, University of Washington, *chairman;* H. H. Douglas, Pres., Southern California Chapter, ASHVE, Air Conditioning and Heating Engineer, Southern California Edison Co.; J. H. Gumz, Pacific Gas & Electric Co.; M. J. Hauan, Pres., Pacific Northwest Chapter, ASHVE, Consulting Engineer; Daniel Hayes, Board of Directors, *Heating, Piping and Air Conditioning Contractors National Association;* J. E. McNevin, Pres., *HPACCNA*, Manager Colorado Heating Company.

N. H. Peterson, Past Pres., Golden Gate Chapter, ASHVE, Manager, Trane Co., San Francisco; T. E. Taylor, Pres., Oregon Chapter, ASHVE, Consulting Mechanical Engineer; Dr. B. M. Woods, Research Committee, ASHVE, Professor of Mechanical Engineering, University of California; and Charles F. Roth, Manager, and E. K. Stevens, Associate Manager, of the Exposition.

Exhibitors

812 ..... Aerofin Corp.
811 ..... Air-Maze Corp.
714 ..... Alco Valve Co.
403-407 ..... Aldrich Co.
D ..... American Air Filter Co., Inc.
411 ..... American Artisan
1 ..... AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
635 ..... Anemostat Corporation of America
821 ..... Atlas Heating & Ventilating Co. Ltd.
934 ..... Basford Co., H. R.
516-518 ..... Bell & Gossett Co.
621 ..... Brown Instrument Co.
424 ..... California Arts & Architecture
Space to be assigned ..... California Division of Mines
815-817 ..... Carrier Corp.
532 & 629 ..... Chicago Pump Co.
816 ..... Cole-Sullivan Engineering Co.
410 ..... Detroit Lubricator Co.
609 ..... Domestic Engineering
718 ..... Dowagiac Steel Furnace Co.
722 ..... Dunham Co., C. A.
909 ..... Edison, Inc., Thomas A., Instrument Division
809 ..... Fairbanks, Morse & Co.
909 ..... Fedders Mfg. Company
934 ..... Fraser Furnace Co.
A ..... Fraser & Johnston Company
714 ..... Friez & Sons, Julien P., Div. of Bendix Aviation Corp.
522-524-526 ..... Fulton Sylphon Co. Inc., The
617 ..... General Controls Company
E and F ..... General Electric Company, Air Conditioning and Commercial Refrigeration Dept.
417 ..... Grinnell Company of the Pacific
926 ..... Heating Equipment Company
411 ..... Heating, Piping & Air Conditioning
814 ..... Illinois Engineering Co.
422 ..... Illinois Testing Laboratories, Inc.
413 ..... Independent Pneumatic Tool Co.
523 ..... Ingersoll Steel & Disc Div., Borg-Warner Corp.
439 ..... Johnson Co., S. T.
906 ..... Johnson Service Co.
909 ..... Kaye & McDonald, Inc.
411 ..... Keeney Publishing Co.
909 ..... Kewanee Boiler Corp.
904 ..... McDonnell & Miller
909 ..... Martin, J. O. & C. U.
915-917 ..... Mercoid Corporation, The
621 ..... Minneapolis-Honeywell Regulator Co.
824 ..... Montgomery Brothers
C ..... Nash Engineering Co.
810 ..... Owens-Corning Fiberglas Corp.
826-828 ..... Pacific Gas and Electric Co.
912-914-916-918 ..... Pacific Gas Radiator Co.
714 ..... Pacific Scientific Co.
726 ..... Penn Electric Switch Co.
921 ..... Peterson Company, Neil H.
736 ..... Randall Graphite Products Corp.
710 ..... Ray Oil Burner Co.
512 ..... Reznor Manufacturing Co.
613 ..... Ric-wiL Company, The
822 ..... Sarco Company, Inc.
515 ..... Skuttle Sales Company
511 ..... Spang Chalfant, Inc.
509 ..... Torrington Mfg. Co.
412 & 509 ..... Trade-Wind Motorfans, Inc.
425 ..... Triplex Heating Specialty Co. Inc.
418 ..... Tuthill Pump Company
421 ..... Tuttle & Bailey, Inc.
922 ..... United States Register Co.
2 ..... Utility Fan Corporation
521 ..... Waterloo Register Co., The
818 ..... Webster Electric Company
510 ..... Webster & Co., Warren
415 ..... Western Blower Co.
B ..... Westinghouse Electric & Mfg. Company
714 ..... White-Rodgers Electric Co., St. Louis, Mo.
517 ..... Young Radiator Company
514 ..... Zink Company, John

PROGRAM

52nd Annual Convention

HEATING, PIPING AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION

Time given is Pacific Standard Time.—All sessions will be in Hotel St. Francis

Sunday, June 15, 1941

10:00 a. m. Meeting of Board of Directors

Monday, June 16, 1941

8:30 a. m. Registration of Members and Guests

10:00 a. m. Joint session with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel St. Francis

- Walter L. Fleisher, President, ASHVE, presiding
- Invocation
- Singing of "America"
- Greetings from San Francisco
- Response by J. E. McNevin, President, *HPACCNA*
- Greetings by Clyde E. Bentley, President, Golden Gate Chapter, ASHVE
- Greetings by Daniel Hayes, President, Heating and Piping Contractors Association of San Francisco
- Program announcements
- Photograph
- Address—

12:15 p. m. Joint luncheon with AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hotel St. Francis

2:00 p. m. Opening of Pacific Heating & Air Conditioning Exposition, Civic Auditorium

7:00 p. m. Annual Dinner and Dance, Hotel St. Francis

Tuesday, June 17, 1941

9:30 a. m. Call to Order

- Report of Credential Committee
- Appointment of Convention Committees
- President's Address—J. E. McNevin
- Convention photograph
- Report of the Treasurer—S. Austin Pope
- Report of Auditing Committee—John N. Nelles
- Report of Boiler Output Committee—Harry M. Hart, Chairman
- Discussion of Report
- Report of Publication Committee—George P. Nachman, Chairman
- Discussion of Report
- Address—
- Report of Committee on Welding—Thomas I. Magan, Chairman
- Discussion of Report
- In Memoriam
- Announcements
- Adjournment

12:30 p. m. Luncheon for Advertisers in *Official Bulletin*
Secretaries' Conference

7:00 p. m. Round Table Discussions

- Committee chairmen will arrange group meetings for those who wish to discuss special subjects at greater length than is possible on convention floor
- Exposition and Sightseeing

Wednesday, June 18, 1941

2:00 p. m. Call to Order

- Report of Membership Committee—William H. Driscoll, General Chairman
- Discussion of Report
- Report of Committee on Standards—A. F. Nass, Chairman
- Discussion of Report
- Report of Committee on Air Conditioning—A. C. Buensod, Chairman
- Discussion of Report
- Report of Secretary—Joseph C. Fitts
- Report of Committee on Apprenticeship—William V. Hoier, Chairman
- Discussion of Report
- Address—(Defense Construction)
- Report of Committee on Trade Promotion—Edwin P. Ambler, Chairman
- Discussion of Report
- Report on Plumbing and Heating Industries Bureau—George H. Dickerson
- Conditions in the Industry—Reports by Benjamin S. Bacon, New York, (Atlantic Seaboard), H. Merwin Porter, Minneapolis (Mississippi Valley), and by J. Moriarty, Los Angeles (Pacific Coast)

Thursday, June 19, 1941

9:00 a. m. Call to Order

- Report on Committee of Ten—Coal and Heating Industries, Harry M. Hart
- Report on American Standards Association—Harry M. Hart
- Report of Committee on Certification—W. M. Smith, Chairman
- Discussion of Report
- Address—"Practical Benefits of the I.B.R. Research Program"—R. E. Ferry, General Manager, The Institute of Boiler and Radiator Manufacturers
- Radiator Heating in the Small Home
- Address—Selling Radiator Heating in the Small Home Field, by Walter H. Oleson, Executive Manager of the *Heating, Piping and Air Conditioning Milwaukee Association*
- Address—Installing Radiator Heating in the Small Home, by J. A. Ljunggren, Chicago, Ill.
- Address—Employer and Employee, by William C. O'Neill, General Organizer of the *United Association of Journeymen, Plumbers and Steam Fitters*
- Report of Convention Committees—Nominating Committee, Resolutions Committee
- Election of Officers
- Installation of Officers
- Unfinished Business
- New Business
- Announcements
- Adjournment

2:00 p. m. Meeting of Board of Directors
Inspection Trip to University of California

7:00 p. m. Exposition

Friday, June 20, 1941

HPACCNA—Joint Program—ASHVE (See p. 399)

PHILADELPHIA NOMINATES OFFICERS

April 10, 1941. The regular annual meeting of the Philadelphia Chapter was held at the Engineers Club and was well attended. The meeting was called to order by Pres. C. B. Eastman and the minutes of the March meeting were read and approved.

Following the report by the treasurer, E. K. Wagner reported that the auditing committee had examined the accounts of the Chapter and found them to be correct.

J. H. Hucker spoke of the work the publicity committee is doing in connection with the Annual Meeting of the Society to be held in Philadelphia in January 1942, and requested the members to submit slogans in competition, for use in the publicity in connection with the 1942 meeting. He mentioned that a suitable award would be given at the May meeting to the member whose slogan was selected.

President Eastman then read the By-Laws, *Article VIII, Sections 3, 4* and *5*, governing the nomination of officers and appointed Messrs. W. F. Smith, W. J. Searle, Jr., and J. H. Hucker as tellers. The list of names proposed for officers for the ensuing year by the nominating committee was then read, and as no additional nominations were made from the floor, it was moved, seconded and carried that the nominations be closed, and the secretary was instructed to cast the ballot for the following: *President*—H. B. Hedges; *First Vice-President*—H. H. Mather; *Second Vice-President*—Edwin Elliot; *Treasurer*—A. C. Caldwell; *Secretary*—R. D. Touton.

President Eastman thanked the chairmen of the various committees for their faithful service rendered during his administration, and advised the members that the funds borrowed for incorporating expenses had been repaid from the profits of the Year Book which was so efficiently edited under the direction of the chairman of the Year Book Committee, P. L. Pryibil.

On motion of A. J. Nesbitt, duly seconded and unanimously carried, a rising vote of thanks was extended to President Eastman for his able guidance of the Chapter during his term of office.

H. B. Hedges, chairman of the membership committee, was called upon by President Eastman to report on the work of his committee, and then the meeting was turned over to H. H. Mather, chairman of the meetings committee, who announced the May golf meeting at Bala Golf Club.

An unusually interesting sound film was shown by the courtesy of the Owens-Corning Fiberglas Co., picturing the manufacture of glass wool as used for insulation purposes, and its development for many uses other than for insulation.

S. R. Arem, Jr., engineer of the Westinghouse Electric and Mfg. Co., spoke on the subject of electrostatic air cleaning. Lantern slides were used to illustrate Mr. Arem's talk, and a practical demonstration was given of the operation by the use of a portable electro-static air cleaner. His address was well received and following a general discussion from the floor the meeting adjourned.

PROFESSOR EASTWOOD VISITS PITTSBURGH

April 16, 1941. The April meeting of the Pittsburgh Chapter was called to order by Pres. E. C. Smyers in the Marine Room of the Hotel Roosevelt. The minutes of the previous meeting were approved as read, and all reports and other business, except the report of the nominating committee, were dispensed with because of the length of the program.

A. F. Nass, chairman of the nominating committee, submitted the following candidates for election at the May meeting: *President*—E. C. Smyers; *Vice-President*—C. M. Humphreys; *Secretary*—T. F. Rockwell; *Treasurer*—L. S. Maehling; *Board of Governors*—F. C. McIntosh, G. G. Waters and H. B. Steggall.

President Smyers then introduced Prof. W. Trinks, head of the department of mechanical engineering at Carnegie Institute of Technology, who visited the Chapter in honor of Prof. E. O. Eastwood, First Vice-President of ASHVE and head of the department of mechanical engineering at the University of Washington, Seattle. Professor Trinks spoke briefly on education and engineering and then the meeting was turned over to Professor Eastwood.

Professor Eastwood first presented the greetings of the Council of the Society and then went into the topic of Engineers in National Defense. He devoted his time mainly to the manufacture of airplanes and the development work which must precede the manufacture. He also spoke at length on the problems of heating and cooling airplanes. The audience showed great interest in Professor Eastwood's address, and upon motion of J. F. Collins, Jr., a vote of thanks was extended the speaker, and the meeting adjourned at 10:05 p. m.

WESTERN NEW YORK ANNOUNCES CANDIDATES

April 14, 1941. The monthly meeting of the Western New York Chapter was held at the Iroquois Gas Corporation Bldg., Buffalo, N. Y., and the minutes of the February meeting were read and approved.

Pres. C. A. Gifford appointed J. J. Landers and W. E. Voisinet on the auditing committee to audit the books of the treasurer. W. R. Heath announced the forthcoming visit of Prof. E. O. Eastwood, Seattle, Wash., first vice-president of the Society, stating that there would probably be an informal luncheon in his honor.

President Gifford then announced the scheduled Council Meeting in Toronto and requested that as many members of the local chapter as possible be present at this meeting.

M. C. Beman reported that the Board of Governors had nominated the following list of members for officers for the year 1941-1942: *President* — W. R. Heath; *First Vice-President* — H. C. Schafer; *Second Vice-President* — S. M. Quackenbush; *Treasurer*—B. C. Candee; *Secretary*—Herman Seelbach, Jr.

C. W. Farrar said that a group was being organized to visit Toronto to present an invitation to the Council of the Society for a Buffalo meeting in 1944.

The speaker of the evening, H. L. Volgenau, Assistant Secretary of Buffalo Chamber of Commerce, was then introduced, and gave the Chapter a very interesting talk entitled *Home Building Gets a Lift.*

ATLANTA MEMBERS MEET

April 4, 1941. The regular monthly meeting of the Atlanta chapter was held at the Biltmore Hotel, and was called to order by Pres. S. W. Boyd, with 13 members and guests in attendance. Following the introduction of various members present, the minutes of the previous meeting were read by L. F. Lawrence, Jr., in the absence of Secretary Koch.

C. L. Templin, in reporting for the membership committee, advised that 18 new members have been brought into the Society since the beginning of the year, and he further stated that the Chapter should continue in its active work in securing new members.

L. F. Kent outlined the business covered at the 47th Annual Meeting of the Society in Kansas City in January. He brought out the fact that the Speakers Bureau of the Society will be changed to the Local Chapters Committee, which will coordinate the work of the chapters throughout the country. He stated that the Chapter Delegates will comprise this Committee. He also mentioned that all chapters were urged to plan their meetings for the coming year and after some discussion from the floor, it was decided that the program committee take this under advisement.

President Boyd then introduced W. S. Shipley, chairman of the board, York Ice Machinery Corp., who gave a very clear picture of the operation of the defense program, particularly as regards to plant production. He urged every member present to do his best in an effort to cooperate with the defense program to keep production on an increase.

The meeting was then turned over to F. L. Laseter, who conducted an open discussion of the heating code which the chapter has for consideration. Inasmuch as the question of the Atlanta Chapter's participation in the writing and handling of this code was taken up at the Kansas City meeting, Mr. Kent read a portion of the minutes of the Chapter Delegates Conference, in which the Chapter was advised that the Society could not lend its official stamp of approval to any code. However, it was urged that all members be of any assistance possible in formulating a suitable code which in its final form would be turned over to the City Officials.

There was considerable discussion from the floor and inasmuch as time did not permit each member to state his reaction and suggestions for possible changes in the code, Mr. Laseter asked that all questions of this nature be sent directly to him. With no further business the meeting adjourned at 10:30 p.m.

LOCATING NATIONAL DEFENSE INDUSTRIES TOPIC AT IOWA

March 11, 1941. The March meeting of the Iowa Chapter was held in the Memorial Union, Ames, Ia., and was attended by 28 members and guests. Pres. R. A. Norman presided, and following dinner it was decided to dispense with the regular business meeting to provide more time to the speakers.

Prof. E. S. Lynch, Economics Dept., Iowa State College, then gave an interesting discussion on the effects of locating National Defense industries at various points throughout the country, presenting facts he has gleaned from his work with government officials in this connection. He showed the similarity between engineers and economists on this industry location problem, and pointed out that Washington's attitude is that the location of the various Defense industries must be considered from the Defense standpoint only, not whether the industry would be a better help to one district than another, trying, however, to cause as little dislocation and after-the-war social problems as possible.

Dayton Brundage of the Chicago, Northwestern Railroad Co., who is in charge of air conditioning equipment on passenger trains, then discussed a number of the problems he has encountered in this work. These problems included humidity, temperature control, distribution of air, training of men to operate the air conditioning unit, etc.

February 11, 1941. The February meeting was held at the Fort Des Moines Hotel, Des Moines, and was attended by approximately 20 members and guests. President Norman opened the meeting, and the minutes of the last meeting were read and approved, followed by several announcements by the secretary.

President Norman reported as official delegate to the 47th Annual Meeting of the Society in Kansas City, first reading a letter from E. K. Campbell, Kansas City, a member of the Council, commenting on the functions of chapter committees.

Then President Norman urged that the Iowa Chapter have two delegates to the Annual Meeting, and that one of these men always be the one to attend the following year in order to insure continuity. For example, he recommended that the president be one delegate and the vice-president, who is also chairman of the program committee, be the second delegate, as he would normally be the president for the following year. It was left that the appointment of the chairman of the meetings committee as the second delegate be considered.

Eugene Olson, Iowa State College, then made some comments on the Heating and Ventilating Conference scheduled for April, and asked for suggestions. During the discussion President Norman pointed out that he was somewhat disappointed in last year's Conference, feeling that enough people were not being reached. It was agreed, however, that the Conference is a good thing and should be continued and that every effort this year would be made to reach those who would benefit most by the Conference, and also to make it as practical as possible.

President Norman then made some comments about chapter meetings and programs, and suggested the possibility of securing Dr. Stevens, president of Grinnell College, to attend one of the Iowa Chapter meetings. He explained that most of the chapter meetings should be as inspirational and educational as possible along heating and ventilating lines, and also stated that it would be worthwhile occasionally to have inspirational speakers in other fields, such as Dr. Stevens.

The meeting then adjourned to join the dinner meeting of the Iowa Engineering Society at which movies of the failure of the Tacoma, Washington, Narrows Bridge were shown.